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Bethesda, MD 20817-5700

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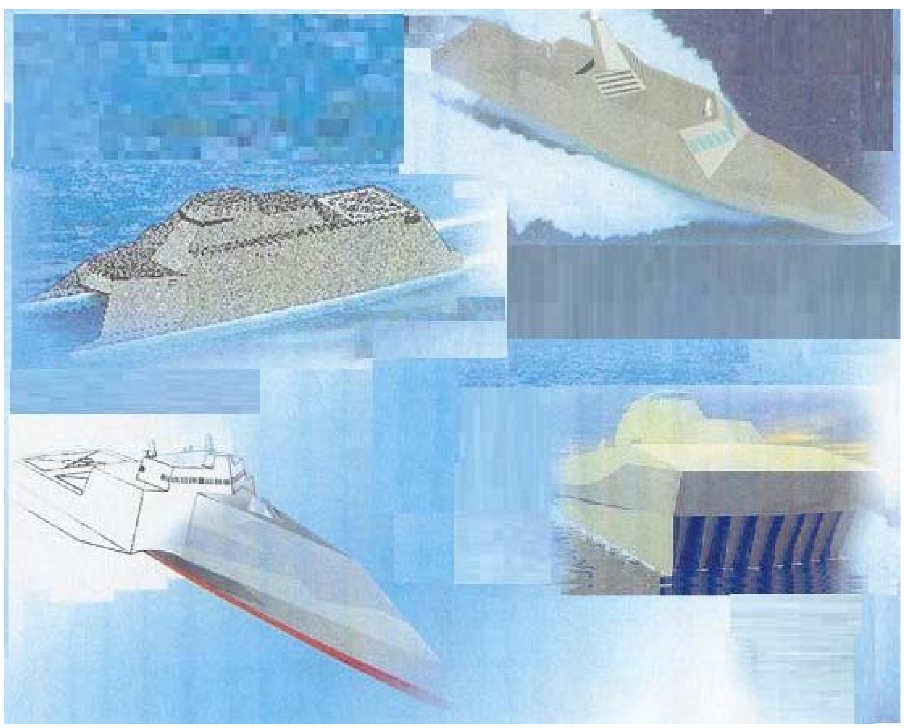
Total Ship Systems Directorate
Technology Projection Report

HIGH-SPEED, SMALL NAVAL VESSEL TECHNOLOGY DEVELOPMENT PLAN

Edited by G. Robert Lamb



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13. ABSTRACT *(Maximum 200 words)*

A high-speed, small Naval vessel innovation cell project was carried out at the Naval Surface Warfare Center, Carderock Division from August through December 2002. The project was chartered by ONR to define the near-term (available in 5 years) technology investments required for 500 to 3,000 ton, high-speed Naval ships. Extensive use was made of technology projections made in 1997 at the High Speed Sealift Technology Workshop held at NSWCCD. Those technology projections were made for: Ship/System concepts, Hullforms, Propulsors, Propulsion Plant, Materials and Ship Structures. This project began by reviewing the high-speed sealift technologies for applicability to high-speed, small Naval ships. Where appropriate, the state-of-the-art data was updated to include recent developments. Spreadsheet parametric models for high-speed monohulls, catamarans and trimarans were used to assess the impact of different technologies on total ship weight and performance. Capabilities needed from each technology were compared with the current state-of-the -art to determine the necessary technology enhancements. Estimates of the development time and cost for each technology were made based on experience with developing similar technologies, engineering estimates, and vendor data. The goal of this plan is to bring the individual technologies to a level of maturity appropriate for ship design and construction.

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EXECUTIVE SUMMARY

The High-Speed, Small Naval Vessel Innovation Cell project conducted at the Naval Surface Warfare Center, Carderock Division from August 2002 through December 2002 had three main objectives:

1. Create a consistent database of existing naval and commercial small high-speed ships in the 500 to 3,000 mt size range;
2. Create spreadsheet parametric models and design relationships for high speed monohull and multi-hull displacement ships, and conduct trade-off studies to assess the impact of different technologies on the total ship weight and performance;
3. Develop a technology roadmap outlining the positive and negative impacts of technologies at the system level, and the technology investments required to achieve small Naval ships with burst speeds of 40 to 60 knots.

This document fulfills the technology roadmap objective. In formulating this Technology Development Plan, extensive use was made of technology projections that had been made in 1997 at the High Speed Sealift Technology Workshop held at NSWCCD. Those technology projections were made in six key areas; namely: Ship/System Concepts, Hullforms and Propulsors, Propulsion Plant, Materials and Ship Structures, and Shipbuilding and Manufacturing. This project began by reviewing the high-speed sealift technologies for applicability to high-speed, small Naval vessels. Where appropriate, the technology state-of-the-art data was updated to reflect recent developments.

Then the revised technology projections for structures and materials, gas turbines, reduction gears, and waterjets were combined selectively into concept ships to examine the whole-ship implications of the technology. Primary focus was given to technologies classed as near-term (available in 5 years). Economic considerations were not introduced at this stage since the initial focus was on determination of technological feasibility and performance enhancement without regard to cost of development or commercial viability.

The capabilities needed from each of the technologies to produce these designs were compared with the technical state-of-the-art for those technologies to define the necessary near-term and far-term technology enhancements. Estimates of the time to develop and rough order of magnitude development costs were made for each of the technologies based on a variety of factors including experience with development of similar technologies, engineering estimates, vendor data, and cost models. The goal of this plan is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction.

This development plan is comprehensive, with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in critical areas is expected to meet anticipated commercial needs for aerospace, industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or

the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military missions. Consequently, the potential existence of these commercial efforts is identified, while the cost reductions that might result have not been shown.

The plans contain some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of far-term SES hulls requires development of SES-specific lift fan and seal technologies. In addition, hullform choice and speed requirement will affect what type of reduction gear and waterjet technology must be developed. Since choices such as these cannot be made with certainty prior to when a commitment is made to specific near-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed.

Year	1	2	3	4	5	Funding ($K)
Hull Form						
- Monohull						6,600
- Trimaran						7,900
- Catamaran						4,400
- Surface Effect Ship						6,100
Hull/Propulsor Integration						8,800
Structures & Materials						23,800
Gas Turbines						20,050
Waterjets						14,000
Reduction Gears						7,500
Lift Fans						2,050
Seals						7,000
Funding ($K)	9,200	35,275	42,075	18,750	2,900	108,200

1.0 INTRODUCTION

The High-Speed, Small Naval Vessels Innovation Cell project was chartered by ONR to define the near-term (available in 5 years) technology investments required to enable development of 500 to 3,000 mt, high-speed Naval ships needed for realistic mission requirements. Specific technology investment selections should be based on detailed design studies, which were beyond the scope of this project. Instead, the objective was to assess whole-ship implications of technology in a generic fashion. The required mix of technologies depends on three mission requirements: speed, range and payload. The design space can be thought of as a three-dimensional box, as shown in Figure 1-1. Assumed design "burst" speeds varied from 40 to 60 knots. Required transit range at economical speed of 18 to 20 knots was 3,000 to 5,000 n.mi. Specific payload items are difficult to model, so payload densities were used instead.

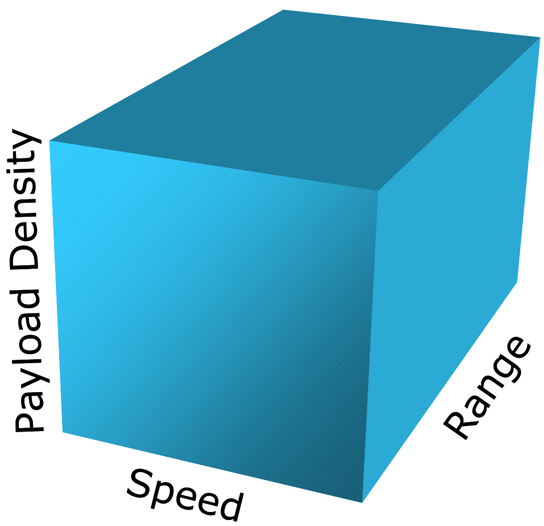

Figure 1-1: Small Naval Vessel Study Design Space

Three generic mission categories were developed. Since future payload items were not known, representative payload weights and volumes for each of the missions were developed for the appropriate range of ship sizes.

- Combatant – Payload packages of sensors, weapons, and guns totaling 53 mt for a ship of about 500 mt and 299 mt for a ship of about 3000 mt.
- Air Operations – Payload of unmanned air vehicles and helicopters totaling 85 mt for a ship of about 500 mt and 128 mt for a ship of about 3000 mt.
- Cargo ship – Payload of material, equipment, and troops totaling 254 mt for a ship of approximately 1500 mt and 606 mt for a ship of about 3000 mt.

Figure 1-2 shows the range of payload weights and volumes considered for the three generic mission categories for ships displacing between 500 and 3,000 mt. Ships smaller than 1,500 mt were judged unsuitable for carrying military cargo. The slope of each line is the payload specific volume (cu. m. per mt), which is the inverse of payload density. As would be expected, the line for Air Operations payload volume has the steepest slope (14.15 m^3 per mt), which means this payload density is lowest. The combatant payload line has the most gradual slope (6.54 m^3 per mt), which translates into the highest density. The slope of the cargo payload line is closer to that for air operations than to the combatant line. Combatant payload densities were used for all of the technology payoff investigations because it is the densest payload. It follows that weight reduction is most beneficial for the high-speed combatant mission.

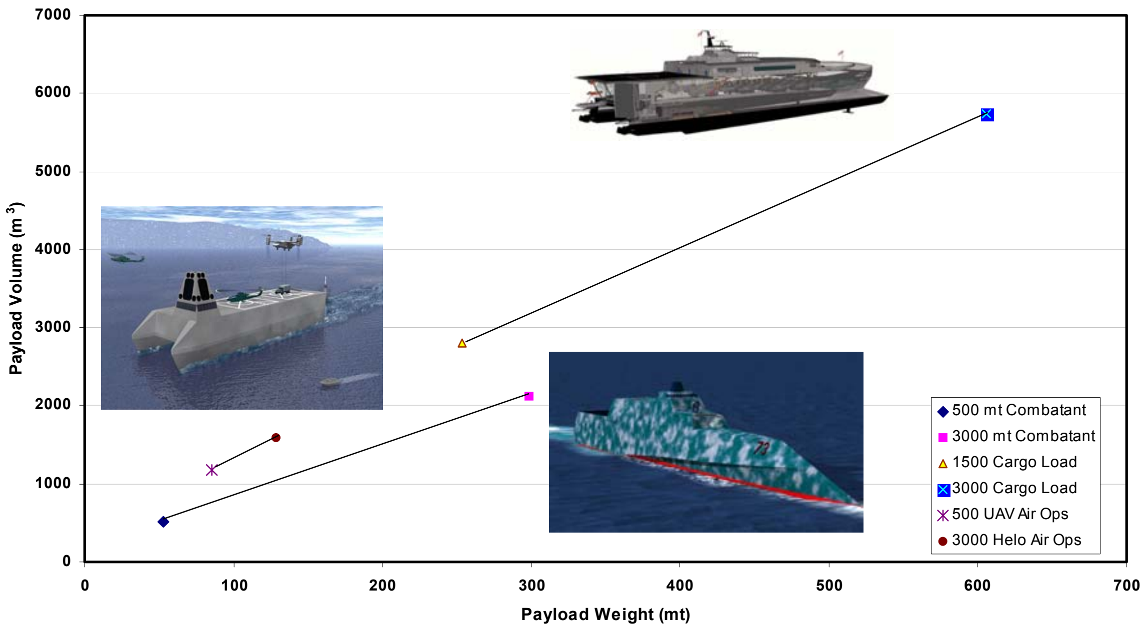

Figure 1-2: High-Speed, Small Naval Vessel Payload Weight and Volume

It was originally envisioned to have one Excel spreadsheet-based, parametric modeling program to handle all three hull forms (monohull, catamaran and trimaran), but that was found to be too technically difficult. Instead, three separate modeling programs were developed.

The trimaran parametric modeling program was the first one developed, because a trimaran sizing spreadsheet had been created by the High Speed Sealift Innovation Cell. The monohull parametric modeling program was basically the trimaran program with the side hulls deleted. The major modules that were modified were structures, hydrodynamics, and powering. The catamaran model was the most extensively modified from the original trimaran parametric model. Modules that had to be modified were structures, powering, machinery, and hydrodynamics. An SES spreadsheet-based modeling program was desired, but it would have required major effort to develop, and was considered to be beyond the scope of this project.

Monohull, trimaran and catamaran designs were produced for the upper and lower bounds of the combatant payload. Development of point designs for each of the hullform types to a uniform

standard was a priority. Common design standards, margins, manning assumptions, and weight algorithms were adopted where practical and appropriate. Updated technology projections for structures and materials, gas turbines, reduction gears, and waterjets were combined with additional technical information to produce a common basis for these technologies in the designs.

The main aim of the Innovation Cell was to derive a Technology Development Plan (TDP) on the basis of demonstrable need and platform performance pay-off. Primary focus was technologies classed as near-term (available in 5 years). There were also potential spin offs from the designs developed including technology pointers as to key factors to particular missions, realistic ship concepts for operational analysis and planning, and creation of a basis for discussions with other organizations and exploration of their interest in research involvement and developing technologies.

The capabilities needed from each of the technologies to produce these designs were compared with the technical state-of-the-art for those technologies to define the necessary near-term technology enhancements. Estimates of the time to develop and rough-order-of-magnitude development costs were made for each of the technologies based on a variety of factors including experience with development of similar technologies, engineering estimates, vendor data, and cost models. The goal of these plans is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction. Technology development plans for each of the technologies are provided in the following sections of this report.

These development plans are comprehensive with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in critical areas is expected to meet anticipated commercial needs. For example, development of large gas turbine technology is highly likely for aerospace, industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military missions. Consequently, the potential existence of these commercial efforts is identified while the cost reductions that might result have not been shown.

The plans that follow contain some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of near-term SES hulls requires development of SES-peculiar lift fan and seal technologies. Since no commitment has been made yet to specific near-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed. Choices between alternatives will likely be made to further focus the technology development effort and reduce cost. Consequently, a representative comprehensive program is summarized in the last section of this report.

References

1. “High-Speed Sealift Technology Development Plan”, Carderock Division Naval Surface Warfare Center Technology Projection Report NSWCCD-20-TR-2002/06, May 2002.

2.0 SHIP HULLFORM CONCEPTS

2.1 Introduction

This Small High-Speed Ship Technology Development Plan is based on design studies produced using three displacement hullforms: monohull, catamaran, and trimaran. Technology needs for one powered lift hullform, the surface effect ship or SES, are also addressed fully. All of these hullforms are considered viable candidates for high-speed ship missions due to the existence of a technology base that is suitable to support design and construction of each hull type, although thus far for lower levels of performance. Proof of concept demonstrations of at least one version of each hullform has also been achieved through full-scale operational experience including numerous commercial variants of these hulls, but for somewhat lower speeds. Naval high-speed (40 to 60 knots) missions require extrapolation beyond current capabilities in critical areas such as structural loads, resistance and powering, and seakeeping. The necessary technology development encompasses model test data, development of analysis tools, and development of design standards and practices such as those required for structural classification.

While the technology extrapolations differ for the different hullforms, the magnitude of the extension is roughly comparable for monohulls, trimarans and SES. The existence of a mature catamaran industry that is producing commercial high-speed ferries with similar characteristics (2,000 to 3,850 mt displacement, 40 knots) indicates that only modest technology evolution is necessary to produce catamaran designs for small HS ship missions.

The near-term HS Naval ship designs envisioned are faster than existing displacement ships. Today's monohull and catamaran fast ferries, with only a few exceptions, have speeds of at most 42 knots. The largest trimaran is the 1,100 mt *Triton*, which has a service speed of only 20 knots. The largest existing SES is smaller than required, weighing about 1,500 mt, but it is capable of high speed – 54 knots. However, a 4,000 mt SES with a service speed of 38 knots is expected to start operating in Japan in 2005. Technical risk in extrapolating the current displacement hulls to meet the more demanding high-speed missions will be reduced significantly through design, construction, and technical validation of intermediate-size high-speed ships which use slender hulls similar to those envisioned for HS roles. While such a progressive approach to evolution of hullform technology is prohibitively expensive if attempted for all of the hulls, it is strongly recommended for any hullform(s) chosen for development.

2.2 Monohull

2.2.1 State-of-the-Art

While the U.S. shipbuilding industry has extensive experience designing and building monohulls of the size required for the HS ship missions, speeds of these ships are generally slower than the 40-60 knots required for HS ship designs. The 72 m Swedish corvette *Visby* shown in Figure 2.2.1-1 is one example of the current state of the art in Europe. This 650 mt, gas turbine powered ship has a top speed of 38 knots and incorporates stealth technology.

Figure 2.2.1-1: 650 mt Swedish Corvette *Visby*

Large, fast monohull ferries ranging between 2,000 and 4,000 mt have been built by shipyards in France, Italy and Spain for routes in the Mediterranean ocean. Typically, they have deep-Vee hullforms and have maximum service speeds of 38 to 42 knots. Figure 2.2.1-2 shows the 134 m long, Corsaire 13000 class ferry built in 2000 by Alstom Leroux Naval in France. This steel-hulled ship displaces about 3,400 mt and has a service speed of 42 knots.

Figure 2.2.1-2: Corsaire 13000 Ferry *NGV Liamone*

Only special purpose designs such as the 1,100 mt Italian built yacht *Destriero* , which crossed the Atlantic at an average speed of 54 knots, have demonstrated significantly higher speeds. Even *Destriero* is much slower, about 43 knots, with a full fuel load.

The size-speed relationship of representative existing monohulls with steel, aluminum and composite hulls is shown in Figure 2.2.1-3. The figure illustrates that a significant increase in speed over current capabilities is required for near-term, small HS ship missions

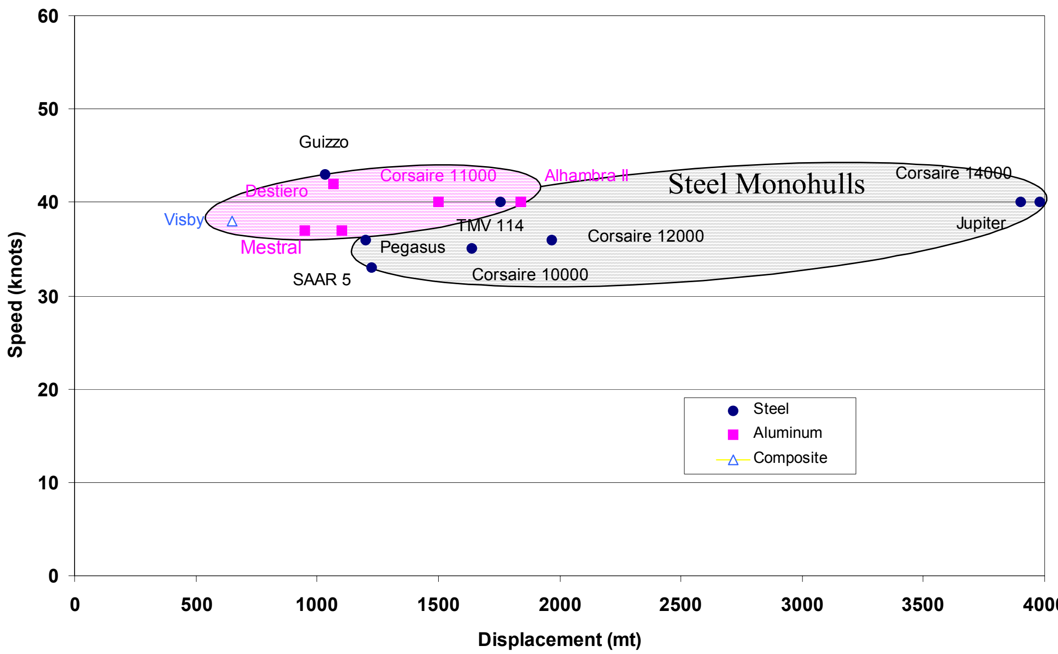

Figure 2.2.1-3: Monohull Technology

Achieving the speed and range requirements identified for near-term small Naval monohulls requires significantly more slender hulls than traditional designs. Limited model testing and analytical investigation of very slender monohulls was carried out by Kvaerner Masa Yards in Finland in 1992 and 1993 as part of their EuroExpress project. In Korea, Hyundai Maritime Research Institute carried out resistance tests on a systematic series of very slender monohulls in 1997. NSWCCD measured the resistance of the very slender center hull of a high speed trimaran design in 2001. Further expansion of the technology base for advanced, very slender monohulls is required to allow reliable prediction of vital design characteristics such as sea induced loads, resistance, powering, seakeeping, and maneuvering. The hydrodynamic integration of high-power waterjets into these slender hulls is of particular importance to minimize installed power, minimize fuel consumption, and assure reliable operation in representative sea conditions. The needed technology includes extension of analytic models and computer programs to address the slender hulls and higher speeds as well as comprehensive model test data.

Slenderness and high speed also have pronounced effects on structural design and performance. Low structural weight is a design priority, but high speeds are likely to result in significant slam

loads in Sea State 5 or worse conditions. Because hull girders for the hydrodynamically slender hulls are also structurally slender, structural loads and reactions to the loads such as slam induced whipping, both vertical and lateral, are expected to be of critical importance. The resulting high-frequency, large-amplitude accelerations are expected to have significant effects on cargo, crew, and hull fatigue life.

2.2.2 Technology Goals

Technology advances for slender high-speed monohulls are needed to reduce the risk associated with ship sizes of 2,000 – 3,000 mt needed to support expected HS Naval missions. Technology development is required in the following areas:

Structural loads – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (covered under section 4.2 Loads).

Resistance and powering – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).

Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 3.2 Powering and section 5.3 Waterjets).

Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/Propulsor Interaction).

Seakeeping – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

Maneuvering, dynamic stability, and control – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under section 3.4 Maneuvering).

2.2.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and hydrodynamic performance of 500 to 3,000 mt slender monohulls operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of high-speed monohulls. Technology development efforts will focus on the development, analysis, and testing of representative slender monohull concepts. The tasks, time to complete each task, and cost associated with developing the needed monohull technology are shown in Figure 2.2.3-1.

High-Speed, Small Naval Vessel Technology Development Plan
Ship Hullforms

Year	1	2	3	4	5	Est. Cost ($K)
Develop Hull Concepts						300
Technology Development						
- Resistance & Powering						
Build powering model						400
Test						500
Extend tools						800
- Loads & Seakeeping						
Build dynamics model						525
Test						1,425
Extend tools						1,200
- Maneuvering & Control						
Test						600
Extend tools						400
Design Validation						450
Funding ($K)	850	2,000	1,525	1,825	400	6,600

Figure 2.2.3-1: Monohull Technology Development Plan

Two stages of hullform development, model testing, and analysis are shown to address variations in hullform expected and evolution of advanced hullform concepts. Costs shown are engineering estimates, based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), and waterjets (section 5.3). Similarity between monohulls and trimaran centerhulls will result in technology developed being applicable to both hull types.

2.3 Catamaran

2.3.1 State-of-the-Art

High-speed aluminum catamarans are widely used as vehicle and passenger ferries. Many designs are in service with displacements ranging from a few hundred tons to about 3,850 mt with speeds of 35-40 knots. Some small catamarans have pushed the speed envelope above 50 knots, although generally only in sheltered waters. Virtually all of the aluminum catamaran ferries larger than 1,000 mt have been designed and built outside the United States. However, in 2003 Nichols Brothers Boat Builders will begin construction of an approximately 1100-mt, 50-knot aluminum catamaran for the Office of Naval Research. Figure 2.3.1-1 is a photo of the 96 m, 38-knot Incat catamaran *Joint Venture*, which displaces about 1,700 mt and was leased by the Navy in 2001.

Figure 2.3.1-1: 96 m Catamaran *Joint Venture*

The largest aluminum catamaran, Stena's HSS 1500, is shown in Figure 2.3.1-2. Three of these 3,850 mt ferries, which transit at 40 knots fully loaded, are in service out of U.K. ports. Range of even the largest high-performance ferries is generally a few (200-400) hundred n. miles.

Figure 2.3.1-2: 126 m Semi-Swath Catamaran *Stena Discovery*

A number of small aluminum catamaran ferries have been built in the U.S. These vessels are generally foreign designs built under license. The fastest is the 53-knot *Patricia Olivia II*, which displaces 202 mt.

In addition, large steel catamarans displacing 3-5,000 tonnes with speeds below 20 knots were designed and built in the U.S. during the late 1960s for Navy missions. Significant technology was developed for these slow, open-ocean ships addressing critical issues such as seakeeping, maneuvering, loads, and structural design. More recently, two classes of SWATH ships (T-AGOS 19 and T-AGOS 23), a specialized variant of the catamaran form, were also built for

Navy missions. The largest steel twin-hull ship is the 11,750 mt *Radisson Diamond*, a SWATH cruise ship built in Finland in 1992. Service speed of this 131 m long ship is 12.5 knots.

The size-speed relationship of representative steel and aluminum catamaran ships up to 4,000 mt is compared in Figure 2.3.1-3. The figure shows that only a moderate increase in speed is required for near-term HS combatant missions.

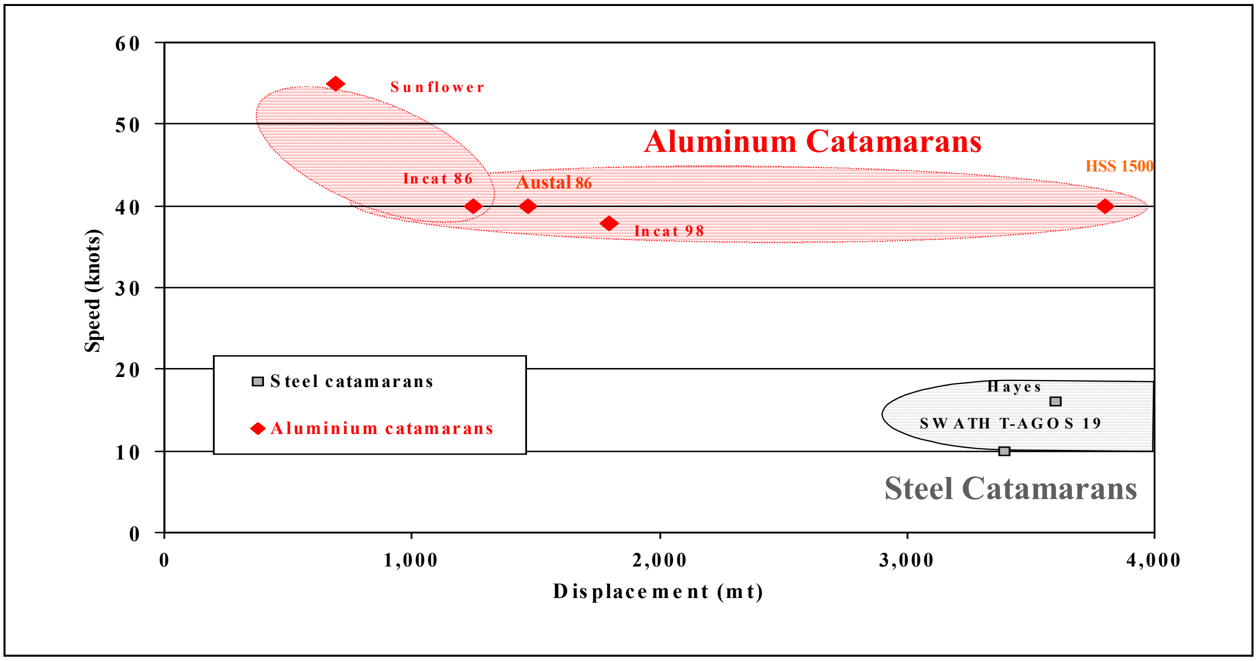

Figure 2.3.1-3: Catamaran Technology

In view of this domestic experience, the existence of a mature international high-speed catamaran industry, and the existence of partnering agreements between U.S. shipyards and foreign catamaran designers/builders assures availability of the catamaran technology needed to build near-term Naval catamarans. Resolution of remaining technical issues such as development of designs to ABS High-Speed Craft Rules at the sizes of interest, completion of training and technology transfer efforts between foreign builders and their U.S. partners, and adaptation of DNV High-Speed Light Craft Rules-based high-speed ferry designs to meet the more stringent military requirements should result from ongoing commercial development.

Although less investment in catamaran technology is needed, a model test program should be undertaken to create a comprehensive database on at least one state-of-the-art catamaran configuration. Little public information is available on the performance of the semi-swath type of catamaran. In addition, improvements are needed in current hydrodynamic analysis and prediction tools, which have significant deficiencies. Beyond that, future catamarans will benefit from generic technology development in structures and materials, gas turbines, reduction gears, and waterjets.

2.3.2 Technology Goals

Technology advances for state-of-the-art, high-speed catamarans are needed to reduce the risk associated with scaling models to the 2,000–3,000 mt displacements needed to support HS Naval missions. Technology development is required in the following areas:

Structural loads – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (covered under section 4.2 Loads).

Resistance and powering – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).

Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 3.2 Powering and section 5.3 Waterjets).

Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/Propulsor Interaction).

Seakeeping – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

Maneuvering, dynamic stability, and control – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under section 3.4 Maneuvering).

2.3.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and hydrodynamic performance of 2,000 to 3,000 mt catamarans operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of high-speed catamarans. Technology development efforts will focus on the development, analysis, and testing of two representative state-of-the-art catamaran concepts, including a semi-Swath. The tasks, time to complete each task, and cost associated with developing the needed catamaran technology are shown in Figure 2.3.3-1.

A single stage of hullform development, model testing, and analysis is thought to be sufficient to fully characterize the technology of a state-of-the-art catamaran hullform. Costs shown are engineering estimates, based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), and waterjets (section 5.3).

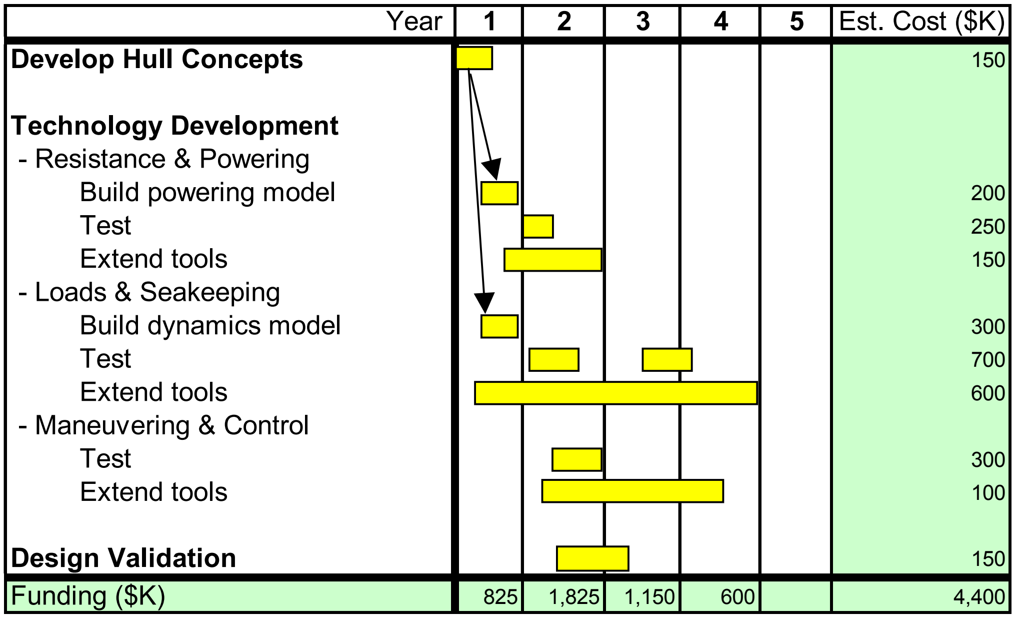

Figure 2.3.3-1: Catamaran Technology Development Plan

2.4 <u>Trimaran</u>

2.4.1 State-of-the-Art

Although a few small trimarans have been built as pleasure craft in the U.S., our domestic shipbuilding industry has little experience designing and building trimarans of the size and speed required for the HS ship missions. The only sizeable high-speed trimaran now in service anywhere is the 40-knot, 250-mt North West Bay ferry *Triumphant.* This aluminum vessel, shown underway in Figure 2.4.1-1, was completed in Australia in 2001.

In addition, technology from model tests, full-scale trials, and design analysis has been produced under the UK/US trimaran joint trials program for the 1,300-mt, 20-knot trimaran RV *Triton*, shown in Figure 2.4.1-2. Built of steel in the U.K., this is the world's first and currently the only trimaran larger than 1,000 mt.

Limited design and model test experience with representative HS trimaran hullforms has also resulted from commercial efforts such as those of BGV Development International, which has carried out model testing and developed designs for 40 to 45 knot trimarans ranging between 600 and 2,000 mt. Kvaerner Masa Marine has also developed designs and carried out resistance optimization studies for trimarans of 30,000 mt with speeds up to 60 knots. This corresponds to a design speed of 41 knots (Froude number = 0.53) when scaled down to 3,000 mt size. Side hull sizes representing 4%, 7%, 10% and 14% of total ship displaced volume (for the pair of

hulls) were investigated. Resistance was predicted for Froude numbers between 0.30 and 0.55. The optimum side hull displacement percentage was found to vary with design Froude number. For Froude numbers above 0.50, values of 8 to 10% were selected.

Figure 2.4.1-1: 54.5 m Trimaran Ferry *Triumpnant*

Figure 2.4.1-2: 98.7 m Trimaran R.V. *Triton*

Figure 2.4.1-3 is an artist's concept of Kvaerner Masa's "slender monohull with outriggers ferry design". Nigel Gee & Associates have explored slender monohull with outrigger designs for ships displacing up to 20,000 mt. In 2002 extensive model testing was carried out on a trimaran with very small outer hulls designed by NSWCCD as a 27,000 mt high-speed sealift ship (see Figure 3.4.1-1). Nominal transit speed of this 323 m long design is 52 knots (Fn = 0.48), but

Figure 2.4.1-3: *Slender Monohull with Outriggers* Ferry Design Concept

resistance was measured up to 100 knots. The two 51 m long outer hulls contributed about 2 % of total ship displacement. The size-speed relationship of HS ship near-term trimarans is compared with representative conventional ships in Figure 2.4.1-4. Unlike the monohull case,

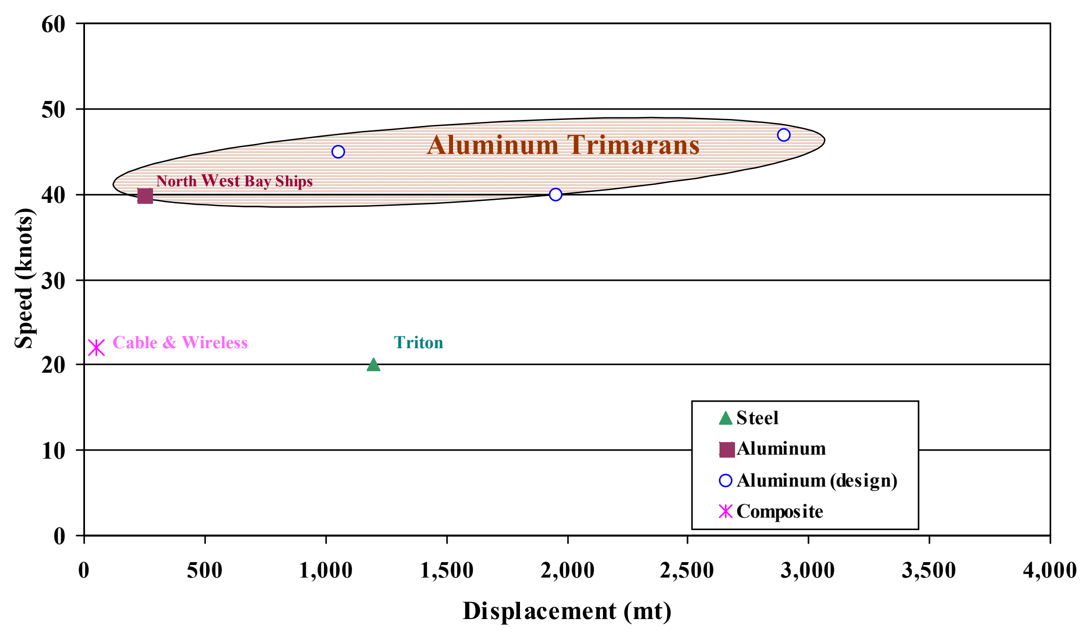

Figure 2.4.1-4: Trimaran Technology

the size and speed requirements for near-term HS Naval trimarans are a significant increase over demonstrated capability.

Most of the Naval HS trimaran hullforms investigated are essentially slender monohulls with small sidehulls added to provide buoyant roll stabilization. The pair of sidehulls typically provide from 2% to 15% of total buoyancy. While the sidehulls add complexity, most technical aspects of trimaran center hulls may be viewed as essentially indistinguishable from the slender monohulls discussed in section 2.2. Consequently, the extensive monohull technology base is also applicable to trimaran center hulls. Similarly, the technology extensions resulting from increased slenderness of HS monohulls are also required for trimarans. Additional trimaran specific extensions are required to address sidehull related issues such as resistance, flow characteristics, seakeeping, loads, structural response, and maneuvering and control of the mainhull-plus-side hulls combination. While these trimaran-specific technology requirements add complexity, just a modest increase in R & D effort overall is required beyond that needed for slender monohulls.

2.4.2 Technology Goals

Technology advances for slender high-speed trimarans are needed to reduce the risk associated with scaling small designs or models to the 2-3,000 mt displacements needed to support HS Naval ship missions. Technology development is required in the following areas:

Structural loads – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (centerhull, sidehull, and cross-structure) (covered under section 4.2 Loads).

Resistance and powering – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).

Propulsion – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 5.3 Waterjets).

Hull/propulsor integration – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/ Propulsor Interaction).

Seakeeping – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).

Maneuvering, dynamic stability, and control – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under section 3.4 Maneuvering)

2.4.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and performance of large slender trimarans operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful designs of these large trimarans. Technology development efforts will focus on the development, analysis, and testing of representative slender trimaran concepts selected to bridge the gap between the hullforms in the current technology base and HS, small Naval ship designs. The tasks, time to complete each task, and cost associated with developing the needed trimaran technology are shown in Figure 2.4.3-1. Two stages of hullform development, model testing, and analysis are shown to address expected variations in hullform and evolution of hullform concepts. Costs shown are estimates based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), loads (section 4.2), structural concepts (section 4.4), ABS HS Naval Craft Guide (section 4.5), and waterjets (section 5.3). Similarity between trimaran centerhulls and slender monohulls will result in technology developed being applicable to both hull types.

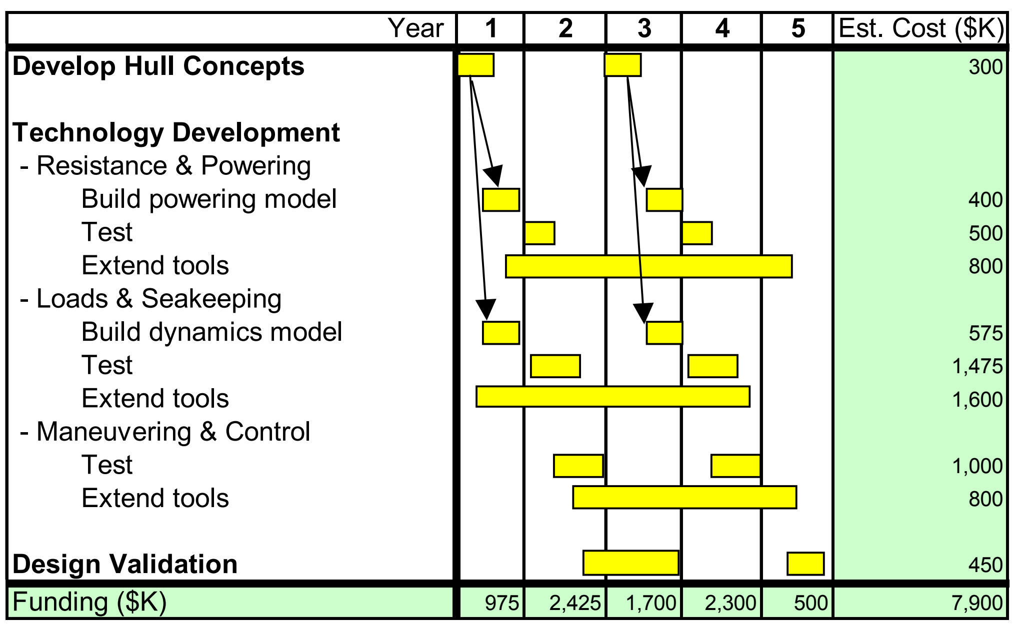

Figure 2.4.3-1: Trimaran Technology Development Plan

2.5 <u>SES</u>

2.5.1 State-of-the-Art

The SES has approximately 40 years of developmental and operational experience in the U.S. and abroad. Several hundred SES have been built and operated. By the late 1970s the U.S. Navy had completed a design and intended to construct a high-speed (80-knot), transoceanic

3,000-ton SES (3KSES) with a cushion length/beam (L/B) ratio of 2.6. This aggressive acquisition program evolved from a technology base that included model tests, analysis, and operation and testing of a series of small manned test craft. Two approximately 100-ton test craft, SES 100A and SES 100B, were built and evaluated in trials at speeds approaching 90 knots to reduce program risk. While the 3KSES program was terminated prior to the construction phase in 1979, a firm SES technology base had been established. Closely related technology was also developed as part of air cushion vehicle (ACV) programs, exemplified by the U.S. Navy's Landing Craft, Air Cushion (LCAC), of which 91 have been built.

In 1978 the 110 Mk1 SES demonstrator was launched and began testing by commercial interests as well as the U.S. Coast Guard. The Navy purchased the SES demonstrator in 1980. In 1981 the U.S. Coast Guard purchased three 152-mt, 30-knot SES 110 *Seabird* class vessels, which were delivered in 1982 and 1983 and operated out of Key West to intercept drug runners. At about this time the Navy converted the SES demonstrator to the 205-mt SES-200 by adding a 15.4 m "plug" section. This resulted in a SES with a L/B of 4.25 and a maximum speed of 28 knots. Subsequently, in 1990, the U.S. Army Corps of Engineers let a contract to modify the SES 200 by replacing the propellers with waterjets and increasing the total propulsion power by 80 percent These modifications increased the craft's maximum speed to 40-knots and extensive trials were carried out with the vessel.

In the mid-1980's the Federal Republic of Germany, in cooperation with the U.S. Navy, undertook detailed development of a 700-mt SES test craft with the highest L/B thus far (4.7) and a speed of 50 knots in calm water, but it was never built. In 1990, the Soviet Union commissioned the largest SES to that time, the 1,000-mt *Dergach*, with a speed of 45 knots. In 1993 the *Oskoy*, the first of nine 370-mt, 20 knot SES craft for mine hunting and minesweeping was delivered to the Royal Norwegian Navy. The SES size boundary was extended further in 1994 when the 54-knot Japanese Techno-Superliner TSL-A70 *Hisho* (renamed *Kibo*) was built, with a displacement of 1,500 mt. The 74 m long *Hisho*/*Kibo* is the largest operational SES. A major new development occurred in Jan. 2003 when a Japanese company signed a contract with Mitsui Engineering & Shipbuilding Co. to build a 140 m SES passenger/cargo ferry with a beam of 29.8 m and a full load displacement of about 4,000 mt. An artist's rendering is shown in Figure 2.5.1-1. This outgrowth of the Techno-Superliner program will have a service speed of

Figure 2.5.1-1: Artist's Rendering of 140 m Long SES Ferry

38 knots and a cruising range of 1,200 n. mi. Design payload is 725 passengers plus 210 mt of cargo. The 140 m SES is scheduled to begin service in 2005. Also noteworthy is the *Skjold*, a stealthy, waterjet propelled 270-mt, 45 knot SES patrol boat built of composites that was completed for the Norwegian Navy in late 1998.

As shown in Figure 2.5.1-2, all of the relatively large SES, except for the 3K SES, have medium to high L/B ratios. A low L/B was chosen for the 3K SES to satisfy the speed goal of 80 knots, but this resulted in high drag at speeds between 15 and 50 knots. Higher L/B ratio SES designs decrease the drag "hump" at moderate speeds, but usually have higher drag than a low L/B SES above 40 knots. Even the high L/B SES 700 design has a substantial drag hump at 12 to 20 knots. While the SES concept is well suited to providing high "burst" speeds in this size range, with the commonly used design approaches it is difficult to also provide substantial transit range capabilities and good seakeeping at moderate speeds.

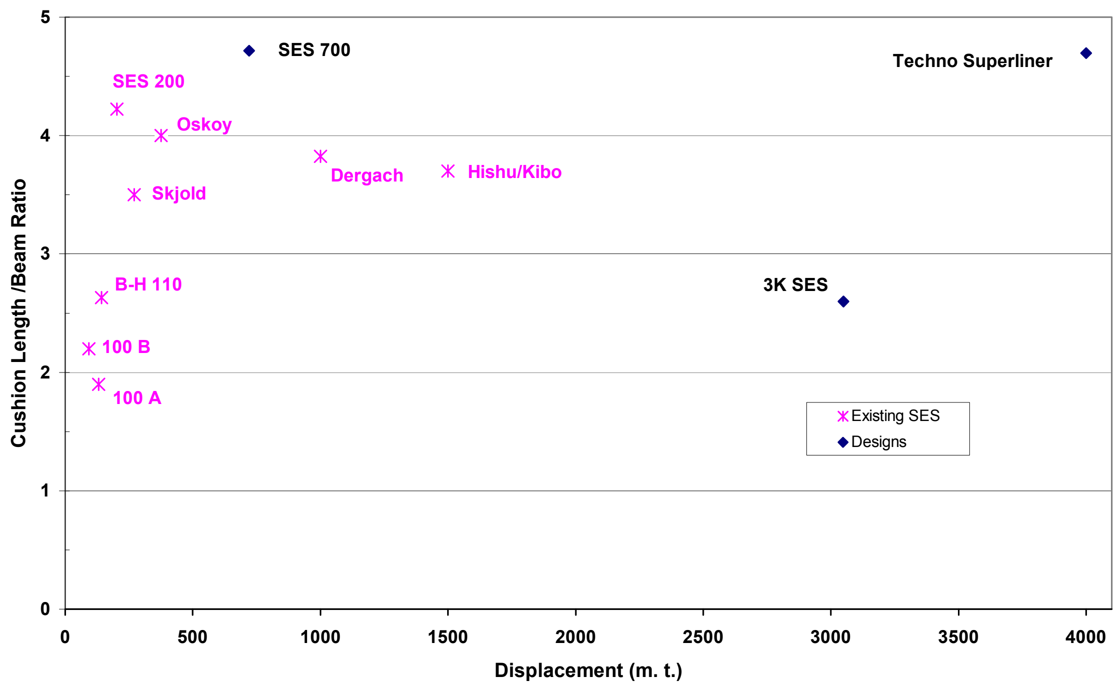

Figure 2.5.1-2: SES Technology

Going "off cushion" or "hullborne" will decrease powering requirements at speeds below 20 knots but in this operating mode the wet deck clearance is greatly reduced and rough sea capabilities will be severely limited. Partial-cushion operation is also possible, which provides intermediate levels of powering and seakeeping performance.

In the early 1980s investigators at NSWCCD carried out theoretical and limited model testing of a surface effect catamaran concept that they called SECAT. In effect, SECAT allows use of a

pair of high L/B cushions within a low L/B vessel planform. These investigations showed that it was more feasible to design a SECAT for low drag over a greater range of speeds. For this reason, SECAT may merit further investigation for small Naval ships with two required operating speeds. Another related concept that has been proposed by a French company is a hybrid trimaran in which a pair of air cushions is separated by a slender displacement type center hull.

2.5.2 Technology Goals

Technology advances are needed for high-speed SES hulls of 1,000 – 3,000 mt that also have reasonable powering performance at 15 to 20 knots to enable long range transits. Twin-cushion or hybrid, cushion-assisted displacement hull concepts appear to have potential but are still in the exploratory stage of development. Technology development is required in the following areas:

> <u>Structural loads</u> – determination of the hydrodynamic forces (primary loads and slamming) and other loads that must be resisted by hull structure (covered under section 4.2 Loads).
>
> <u>Resistance and powering</u> – determination of total resistance due to friction, wavemaking, form drag, etc., added resistance in waves, and the total installed power required to attain a specified speed in specified sea conditions (covered under section 3.2 Powering).
>
> <u>Propulsion</u> – development of waterjet propulsors to provide the thrust needed to attain required speeds (covered under section 5.3 Waterjets).
>
> <u>Hull/propulsor integration</u> – hydrodynamic integration of waterjets and hulls to minimize power and assure reliable seaway performance (covered under section 3.2.3 Hull/Propulsor Interaction).
>
> <u>Seakeeping</u> – analysis of seaway-induced ship motions and their effect on ship and crew performance (covered under section 3.3 Seakeeping).
>
> <u>Maneuvering, dynamic stability, and control</u> – analysis of turning capability, stability in turns, and dynamic control at high speed (covered under section 3.4 Maneuvering).

Additional SES-specific systems level technology development is addressed in section 5.5 SES Lift Fans and 5.6 SES Seals.

2.5.3 Overview of Development Plan

Technology development will be required to characterize the structural loads and hydrodynamic performance of large, high-L/B SES operating at high speed in rough water. Test data will be used to extend and validate analytical design tools and predictive methods, support development of classification standards, and increase confidence in the capability to produce successful

designs of these large SES. Technology development efforts will focus on the development, analysis, and testing of two representative high-L/B SES designs selected to bridge the gap between the hullforms in the current technology base and HS ship hulls. The tasks, time to complete each task, and cost associated with developing the needed SES technology are shown in Figure 2.5.3-1. Costs shown are estimates based on expected scope of testing and facilities required. This hullform specific program will provide essential data to other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), maneuvering (section 3.4), loads (section 4.2), structural concepts (section 4.4), ABS HSS Guide (section 4.5), waterjets (section 5.3), lift fans (section 5.5), and seals (section 5.6).

Year	1	2	3	4	5	Est. Cost ($K)
Technology Development						
- Resistance & Powering						900
- Loads						1,900
- Maneuvering & Dynamic Stability						800
- Seakeeping						2,500
Funding ($K)	825	2,100	2,500	675	0	6,100

Figure 2.5.3-1: SES Technology Development Plan

3.0 HYDRODYNAMICS

3.1 Introduction

A variety of hullforms potentially can be used to satisfy mission speed, range and other requirements for naval ships with full load weights between 500 and 3,000 mt. For purposes of discussing the current state of the art and technology needs it is helpful to identify areas of commonality between the hullform types as well as key differences. Most fundamental is the method of supporting most of the ship's weight at high speeds on the ocean surface:

(a) buoyancy
(b) aerostatic lift, or
(c) hydrodynamic lift.

Speed-power performance of hydrodynamic lift vessels (hydrofoils and planing boats) is addressed in this section for the sake of completeness. Vessels relying on hydrodynamic lift are rarely as large as 500 mt.

Buoyant Support. The great majority of surface ships operating today are supported by the buoyancy of the water displaced by their hull(s). Most of these displacement ships, particularly those designed for moderate speeds, are monohulls. Twin-hull ships, or catamarans, are widely used for fast ferries. Consequently, extensive technology bases currently exist for relatively high-speed monohulls and catamarans. Research and development efforts on high speed trimarans are relatively recent, so the technology base is relatively small.

Aerostatic Lift. Air cushion vehicles, or hovercraft, are wholly supported by a cushion of air supplied by fans and held in by flexible fabric skirts, or seals. ACVs are judged not to be a valid design option for open ocean capable naval vessels in excess of 500 mt because, when operating over water, ACVs have high fan power requirements due to air leakage.

Surface effect ships, or SES, differ from pure hovercraft in having rigid sidehulls. As a result, an SES has less seal leakage and markedly less fan power is required to maintain their air cushion. The first Navy testcraft had thin sidehulls and were designed to operate on cushion most of the time. Beginning with the BH-110, built in 1978, wider sidehulls have been adopted. These provide sufficient buoyancy to support all of the craft's weight, as well as keep the cross-structure out of the water, when the fans are turned off. In this operating mode, an SES is essentially a catamaran with a relatively small amount of wet deck clearance. All oceangoing SES are now designed with fully buoyant sidehulls. Moreover, even when operating at design on-cushion draft, a non-negligible part of the weight of a modern SES is supported by sidehull buoyancy. For example, the German Navy's SES-700 design had 83% cushion support and 17% buoyant support. From this standpoint a modern SES has evolved into a hybrid ship concept. The 4,000 mt. Japanese Techno Superliner SES scheduled to begin service in 2005 will have greater than 25% buoyant support. The hydrodynamics technology base for SES is extensive, but there are areas where further technology development is needed. This is partly due to the wide range of possible SES hull form configurations.

Hydrodynamic Lift. Planing boats are supported largely by hydrodynamic lift at high speeds, and have a mature technology base. In fact, planing boats are by far the most numerous type of high-speed surface craft. They are the dominant hullform for small military patrol boats as well as recreational vessels. Most planing vessels are considerably smaller than 500 mt. Probably the largest planing vessel is the 1070-mt aluminum hulled *Destriero*, which is 67.7 m long and successfully crossed the Atlantic ocean in 1992 at an average speed of 53.1 knots. Hydrofoils also have a quite mature technology base except for those with catamaran hulls, a fairly recent development. The largest hydrofoil built to date is the US Navy's Plainview, which had a full load weight of 320 mt. However, designs exist for naval hydrofoils as large as 2,000 mt.

3.2 Powering

The need for "burst" speeds, which could be as high as 60 knots, is one requirement shared by many of the designs that are being evaluated for future small naval ships. In order to provide a ship capable of such speeds, the designer must place priority on high-speed efficiency when selecting the hull form and propulsion system. The importance of efficiency is magnified by the weight implications of the large quantities of fuel consumed at high speeds. Hull form selection for a burst speed requirement is relatively well understood.

A major complication is that most naval missions also require good powering performance and fuel efficiency at moderate transit speeds (15 to 20 knots) to provide a substantial range. Not only must the selected hull form have good resistance characteristics at both moderate and high speed but, in addition, the propulsors must have reasonably good efficiency in both speed regimes. It is, therefore, of critical importance that powering estimates and speed predictions be accurate for each type of high-speed hull form being evaluated.

The purpose of this effort is to extend resistance and powering prediction techniques to address the most efficient hullforms and provide a validated basis for sizing and selecting appropriate propulsion systems. A major objective is to validate these analytic models as design tools to support development of superior high-speed naval ships.

The approach to be used to develop powering technology will be based on the following:

- develop hull designs that meet representative requirements using existing data and state-of-the-art analytical tools (e.g. Computation Fluid Dynamics methods).
- predict ship resistance and powering performance and flow about the hulls with appropriate analytic and empirical tools.
- plan and conduct tow tank tests to verify predictions. The models will be designed to represent hull geometries and waterjet propulsors appropriate to HS small Naval ship missions and will be tested for a range of operating conditions, speeds, and sea states.
- correlate test data with predictions to extend and validate predictive techniques.

3.2.1 State-of-the-Art

3.2.1.1 Displacement Ships

Resistance estimates for high-speed monohull and catamaran designs have relied heavily on systematic series model test data such as Series 64, the NPL High Speed Round Bilge Series, and the NSMB High Speed Displacement Hull series. Series 64 consists of resistance data for a family of 27 models divided into the three principal groups shown below. NSWCCD carried out resistance tests on these models in the early 1960s (Yeh, 1965). All of the models are rounded bilge, high- speed forms with relatively large transom sterns. Resistance was measured at Froude numbers (V/√gL) up to 1.50. The models have slenderness ratio (Hull Length/Displaced Vol.$^{1/3}$) between 8.0 and 12.4. Hullforms with a S.R of 12.4 are considered very slender. In general, those hullforms in the intermediates shape group are of most interest for small HS Naval ships because of the need to fit propulsion machinery into the hulls. Measured resistance data for the Series 64 hullforms was used as the basis for this project's spreadsheet estimates of required propulsion power for the various ship sizing studies that were carried out to evaluate technology tradeoffs.

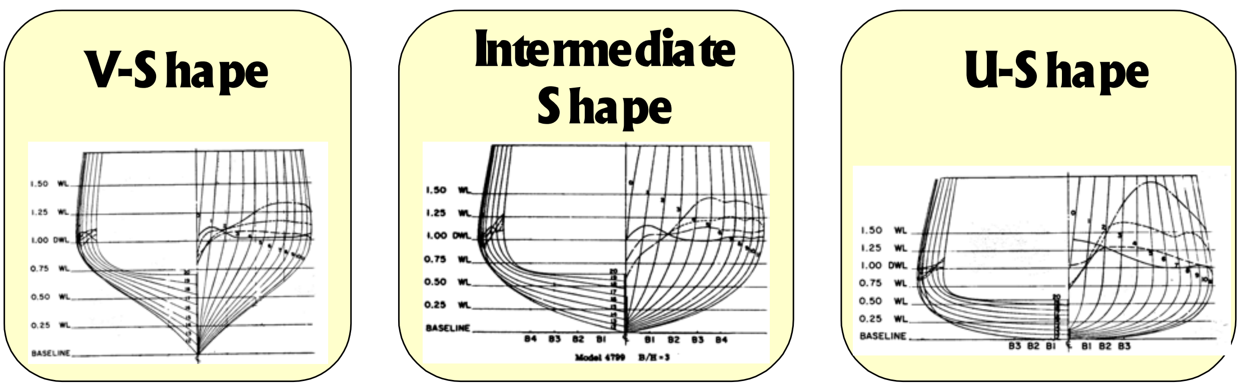

Figure 3.2.1-1: The Three Main Subgroups of Series 64 Hullforms

The actual hullform geometry of many modern high speed monohull vessels differ from these standard series hulls in several ways (section shapes, transom size, bow shape, etc.). Tabulated model resistance data on a family of modern deep Vee hullforms was presented by Grigoropoulos at the annual meeting of The Society of Naval Architects and Marine Engineers in Sept. 2002. These deep Vee monohull forms are similar to those for existing European high speed ferries between 500 and 4,000 mt. Resistance data for the most slender of the tested deep Vee hullforms was scaled up to ship displacements of 750, 1500 and 3,000 mt. Predicted total resistance for these three ship sizes over the speed range from 16 to 60 knots is compared in Figure 3.2.1-2.

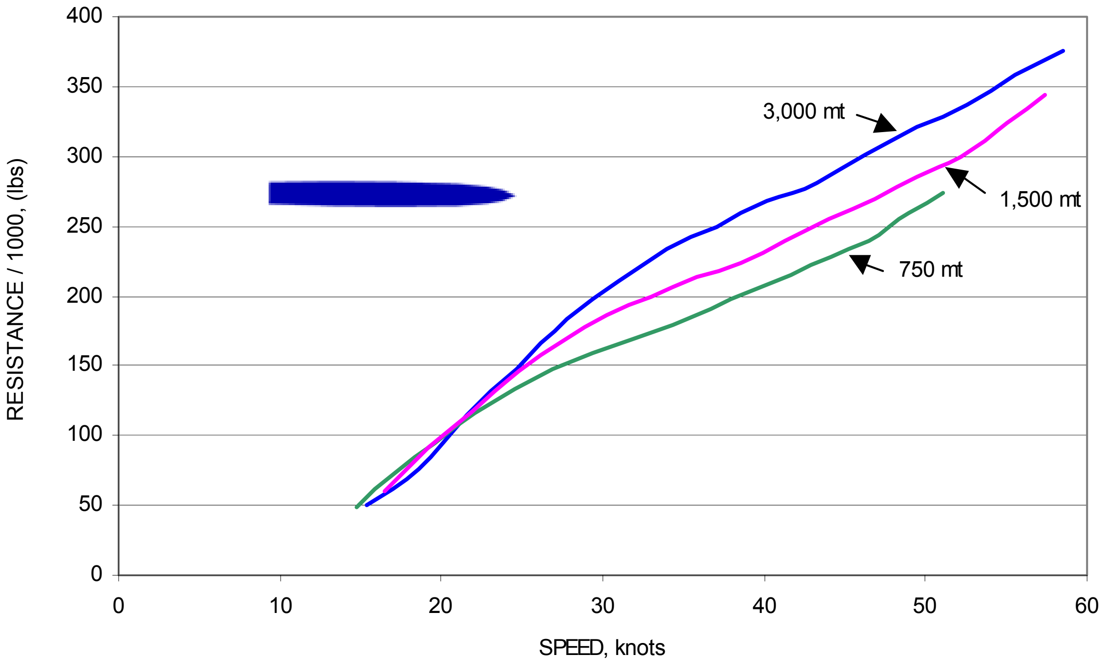

Figure 3.2.1-2: Comparison of Total Resistance for a Deep Vee Monohull Scaled to Three Ship Sizes

While the 3,000 mt ship has the highest resistance at all speeds above 25 knots, as would be expected, the resistance of the 1,500-mt ship is only 10% lower. This comparison illustrates the fact that larger ships are usually more efficient in terms of powering performance. It follows that the required power **per ton of displacement** to drive a 3,000-mt ship at 40 knots will be significantly lower than for a 1,500 or 750-mt ship with the same hullform. Conversely, the smaller ship requires significantly more installed power per ton of displacement to go 40 knots than the 3,000-mt ship, as is shown in Figure 3.2.1-3. Consequently, the designer of a small high-speed ship must allocate a higher fraction of the ship's total weight and volume to propulsion systems.

The deep Vee monohull chosen for these comparisons has a Slenderness Ratio value of 7.7. This is similar to the slenderness ratio values of traditional monohull warships. In the quest for higher speeds there has been growing interest in very slender monohulls. In 1992 and 1993 Kvaerner Masa Yards, in Finland, carried out limited resistance model testing and other investigations of monohull designs with slenderness ratio values as high as 12. Hyundai Maritime Research Institute, in Korea, conducted resistance tests on a systematic series of very slender monohulls in 1997. NSWCCD published a report in 2002 on measured resistance of a model of a 26,600 mt monohull with a slenderness ratio of 10.8. Speeds tested ranged from 25 to 100 knots. Scaled down to a 3,000 mt ship, the speed range tested was 12 to 48 knots.

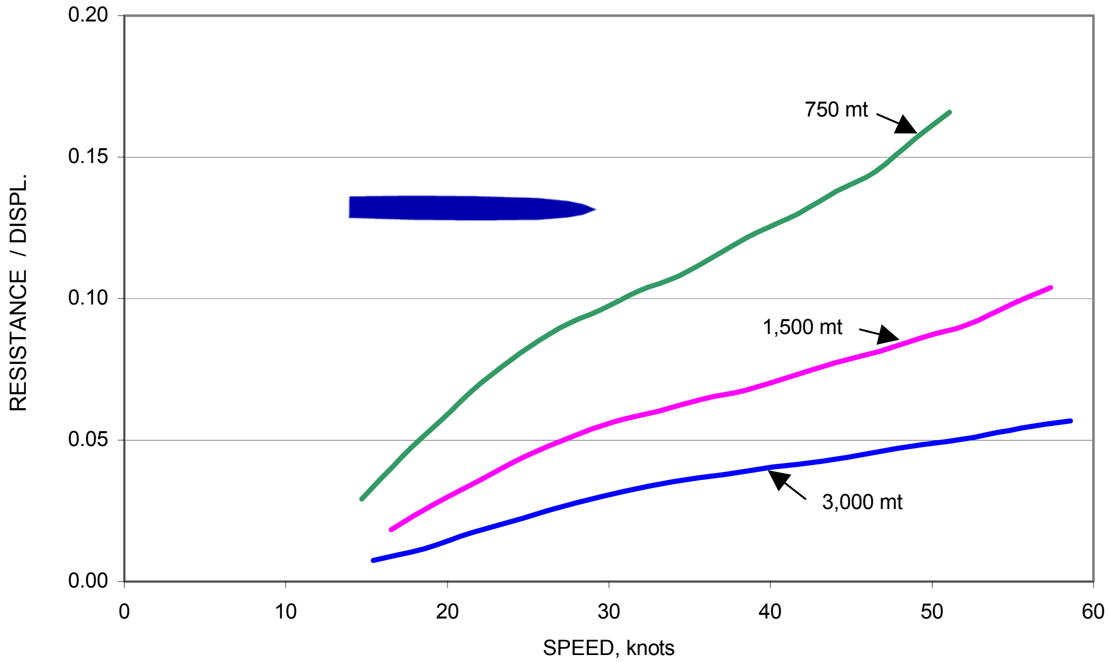

Figure 3.2.1-3: Comparison of Resistance per Ton for a Deep Vee Monohull Scaled to Three Ship Sizes

Over the past 5 or 6 years researchers have published model test data for a couple of systematic series for fairly representative catamaran hullforms (Molland, 1995 and 1997). In many cases, resistance tests were carried out for the actual catamaran and trimaran ships now in service, but this data is proprietary and not generally available. Trimaran configurations have a large number of parameters, so a systematic model test series is problematic. However, model resistance data was published by NSWCCD in 2002 for a 27,000 mt Trimaran design using the center hull with a slenderness ratio of 10.8 and two small side hulls (see Figure 3.4.1-1).

Based on the available model test data, Figure 3.2.1-4 shows a comparison of full-scale effective power requirements for the NSWCCD trimaran scaled to 1,500 mt, and a representative catamaran and deep Vee monohull of the same displacement. Waterline length of the monohull is about 13 % longer than that of the catamaran, but the trimaran is 55% longer. This trimaran has markedly lower resistance than the monohull or catamaran over the speed range between 20 and 45 knots. At 50 knots the monohull and trimaran require about the same power, while the catamaran is noticeably less efficient. The catamaran chosen for Figure 3.2.1-4 had hulls with a slenderness ratio of 8.5, similar to that for the 53-knot, 200 mt *Patricia Olivia II*. It is likely that a different catamaran hullform could be designed that would have somewhat lower resistance at 50 knots. No published resistance data could be found for semi-swath hullforms, which have been utilized for six ferries, including three 40-knot, 3,850-mt ferries operating out of U.K. ports. The semi-swath hullform povides reduced motions compared with a conventional catamaran in exchange for a small increase in wetted surface area and frictional resistance.

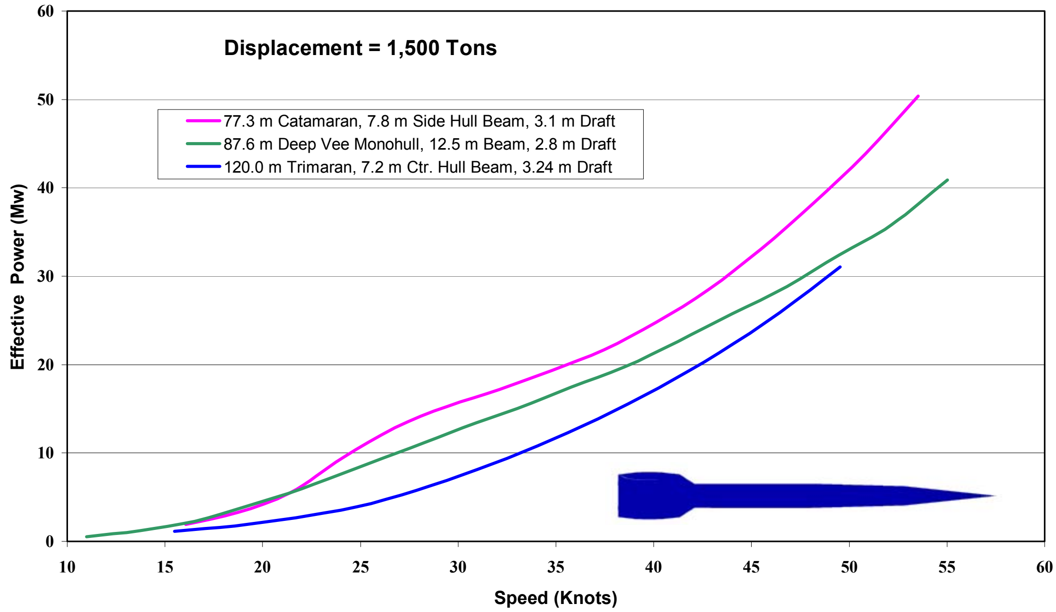

Figure 3.2.1-4: Measured Resistance for Three Types of Displacement Ships

Figure 3.2.1-5 compares the measured resistance of the NSWCCD trimaran scaled to 1,500 mt to the measured resistance of the center hull alone. Even though the sidehulls of this trimaran constitute only 2% of total ship buoyancy, the resistance of the trimaran is noticeably higher than that for the center hull alone above 25 knots. Some of the increase is due to disproportionately high model-scale spray drag from water hitting the underside of the bridging structure.

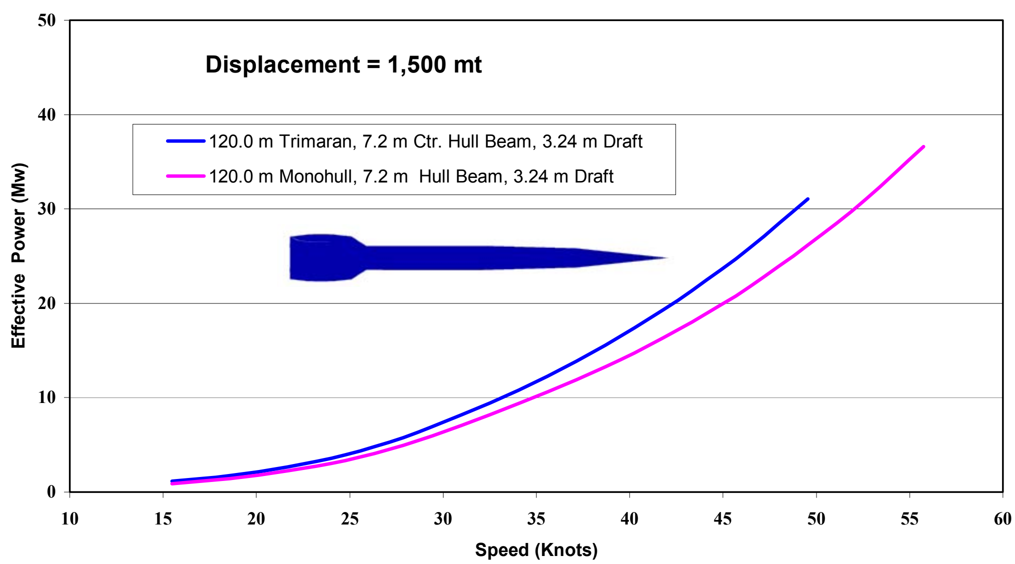

Figure 3.2.1-5: Measured Effective Power for Scaled-Down NSWCCD Trimaran and Its Center Hull

It can be useful, when comparing displacement ships with other ship types, to present resistance characteristics in terms of the ratio of lift to drag. This non-dimensional number can be obtained simply by inverting the calculated resistance per ton of displacement at each speed. Curves of lift/drag ratio as a function of speed are shown in Figure 3.2.1-6 for the NSWCCD trimaran scaled to 1,500 mt and for the corresponding center hull alone. The lift/drag trends for the two hullforms are similar, showing decreasing values as speed increases. The values are also quite high, since an efficient (high aspect ratio) hydrofoil has a lift/drag ratio of 14 to 16. The 1,500 mt slender monohull has a L/D values above 20 at all speeds up to 40 knots, and the values for the trimaran are only slightly lower. At the transit speed of 20 knots the hydrodynamic efficiency of the trimaran is evident from the value of about 70 for its lift-drag ratio.

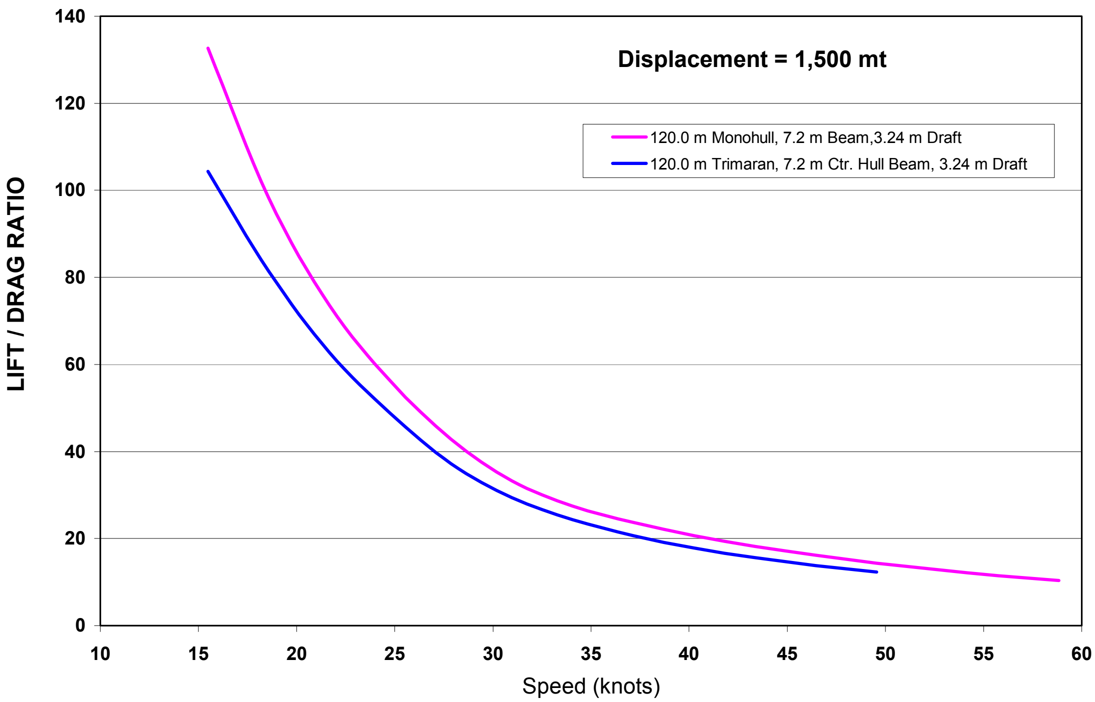

Figure 3.2.1-6: Measured Lift/Drag Ratio for NSWCCD Trimaran and Its Center Hull

Currently, there is no comprehensive publicly available systematic model data base to support development of monohull or trimaran designs employing such slender hulls. Also lacking is published model test data on semi-swath type catamarans. Another gap in the model database is measured resistance of trimarans utilizing one hull of a semi-swath as the center hull. This alternative trimaran configuration has been proposed by several researchers as potentially providing a better balance between resistance and seakeeping performance for small ships.

Ship designers can use advanced panel code computational techniques to predict the resistance of unusual hulls and associated local flow characteristics (waterjet inlets, transoms, streamlines over hulls, nose bulbs). However, these analytical methods require careful correlation with physical data to assure accuracy. Absence of this data is a severe obstacle to the development of mission-specific designs and also hinders generic high-speed hull research and design tool

development. Also needed by designers is a relatively simple analytic resistance prediction tool which can handle different combinations of simple component hullform shapes to enable early-stage design tradeoffs.

3.2.1.2 Surface Effect Ships

The surface effect ship (SES) concept was pioneered by the US Navy in the 1960s. SES are different from pure hovercraft because they have rigid sidehulls. Generally, flexible fabric bow and stern skirts, or seals, are employed to hold in the cushion of air generated by lift fans. The SES concept seeks to obtain a favorable lift/drag ratio by expending a relatively small amount of fan power to lift most of the hull out of the water, thereby markedly decreasing both frictional and wavemaking resistance at high speeds. Early US Navy manned testcraft such as the 93.5-mt, 90-knot SES 100B had relatively thin sidehulls. However, wider sidehulls are now preferred. Current SES have sidehulls designed to support more than 100% of the vessel's weight when lift fans are turned off. Such full displacement sidehulls allow an SES to operate as a relatively deep draft catamaran at low speeds. This is commonly termed "hullborne mode" operation.

The trend towards larger sidehulls was carried further with the German Navy's SES 700 design, which was model tested extensively at NSWCCD in the late 1980s. The sidehulls were sized to support almost 17% of the vessel's 720-mt fully loaded weight when cushionborne. A maximum of 83% of the weight would be supported by the air cushion. The SES 700 design has a cushion length/beam ratio of 4.7 and is 59.5 m long. Predicted maximum speed from a pair of waterjets is over 50 knots in calm water. Total propulsion power is 22.0 Mw. In addition, the design had a total of 4 lift fans driven by two 2.0 Mw diesel engines. The 4,000 mt. Japanese Techno Superliner SES scheduled to begin service in 2005 will have greater than 25% buoyant support when cushionborne..

Figure 3.2.1-7 shows semi-quantitatively the predicted variation in total resistance over a range of speeds for the SES 700 for the two limiting operating modes: cushionborne and hullborne. Estimated resistance in calm water is shown. Above 18 knots, total resistance cushionborne is lower than hullborne mode resistance. At speeds below 18 knots, on the other hand, there is a significant hump in the cushionborne resistance curve, due to wavemaking, so there is significantly lower total resistance hullborne. This design had retractable bow and stern skirts to minimize resistance when hullborne. These are just the extremes of operation. Intermediate "partial cushion" operation is also feasible, because the amount of fan lift can be varied quickly as desired from 0 to 100% of the design value to optimize the performance for each unique operating condition.

Tools to predict SES resistance and powering requirements have been developed over the past forty years through a combination of model tests, manned test-craft trials and analytical models. Through 1979, the major thrust of the SES effort was directed towards high speed (60-100 knots), low length-to-beam (L/B) ratio SES. Following the termination of the 3KSES program in 1979, the U.S. Navy redirected the SES studies to higher L/B ratios and slower speeds (e.g. 25-55 knots). A comparison of the required propulsion power for two 1000 mt SES, one with a low

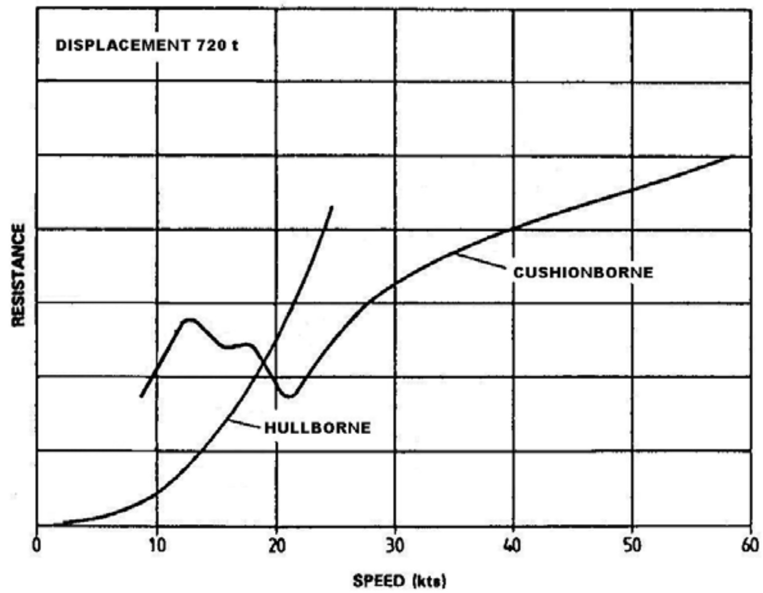

Figure 3.2.1-7: Predicted SES 700 Resistance Hullborne and Cushionborne in Calm Water

L/B cushion and the other with a high L/B cushion, is presented in Figure 3.2.1-8. A similar comparison for 2,000 mt SES is shown in Figure 3.2.1-9. At the 1,000 mt size, an SES with an L/B of 2.5 is more efficient than an SES with L/B of 6.5 at speeds above 51 knots. However, at lower speeds the low L/B SES requires more power. At the 2,000 mt size the cross-over speed increases to about 57 knots. Consequently, in the 2,000 to 3,000 mt size range, SES with a relatively high L/B cushion ratio are the most efficient. However, for SES designs with L/B above 5 it is difficult to provide sufficient cushion depth for good seakeeping while ensuring adequate transverse stability.

Existing analytical models can predict SES performance with sufficient accuracy to support the design of 2,000 to 3,000 mt ships. However, application of CFD tools to hull/propulsor integration is as challenging for SES hulls as for displacement hulls. Consequently, a similar need exists for test data for high-L/B SES hulls at the 40 to 60-knot speeds of interest.

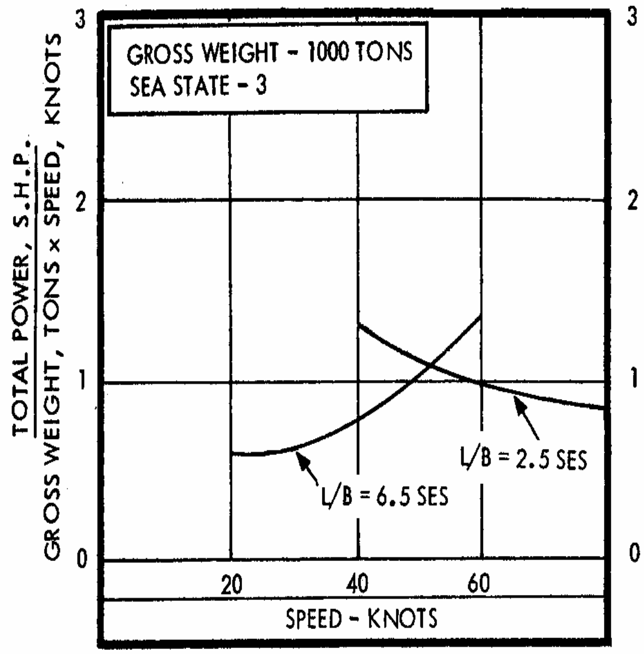

Figure 3.2.1-8: Effect of Cushion Length-Beam Ratio on Required Propulsion Power per Ton Cushionborne for 1000-Ton SES (Eggington, 1975)

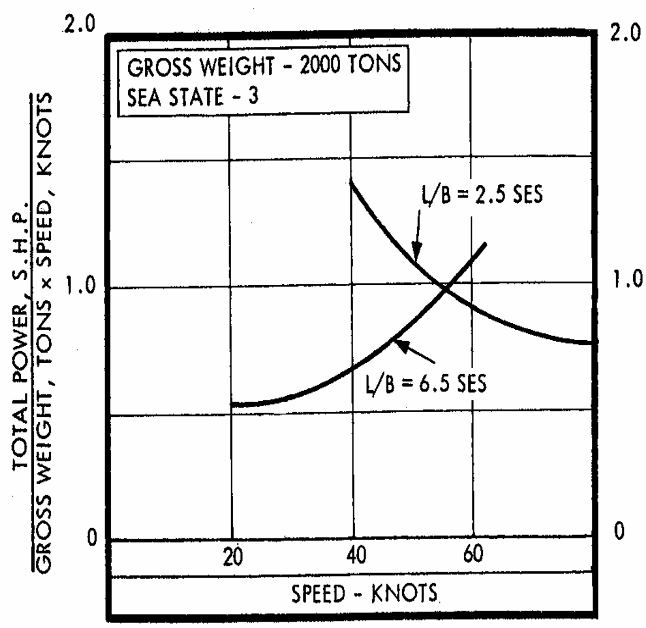

Figure 3.2.1-9: Effect of Cushion Length-Beam Ratio on Required Propulsion Power per Ton Cushionborne for 2000-Ton SES (Eggington, 1975)

3.2.1.3 Hydrodynamic Lift Vessels

Two different ship concepts rely on hydrodynamic lift: **hydrofoils** and **planing boats**. A pure hydrofoil vessel is almost totally supported by hydrodynamic lift. Usually, a pair of large wing-like lifting surfaces (i.e., hydrofoils) is utilized to completely lift the hull above the water surface. Hydrofoils are an old concept, dating back to the late 1880s in the U.S. There are two general hydrofoil configurations: fully submerged and surface-piercing.

With fully submerged hydrofoil configurations the foils are designed to operate below the water surface at all times. A submerged foil system effectively decouples the ship from the surface waves and provides an impressive ability to operate at high speeds in quite rough sea conditions. For example, the Navy's six 241-mt PHM class vessels demonstrated the ability to operate at 40 knots in sea state 5 conditions. Design maximum wave height was 5.0 m. Propulsion was by a pair of waterjets, driven by one LM2500 rated at 13.4 Mw. Maximum speed was 48 knots. Total foil area was 41.2 sq. m. Generally, a fully submerged hydrofoil will become foilborne at approximately 50% of its design foilborne speed and will operate well in rough water over the upper third of its speed range. When a hydrofoil becomes foilborne, there is a marked decrease in their resistance. Consequently, there is a hump in their speed-power curve.

Fully submerged hydrofoil systems are not self-stabilizing. Means must be provided to vary the effective angle of attack of the foils to change the lifting force as ship speed and weight change. Adjustments are also necessary to counter the continually varying apparent angle of attack of the foils in rough seas. This is generally provided by angular changes of trailing edge flaps driven by hydraulic actuators and controlled by an automatic control system.

The AGEH-1 *Plainview*, a 320-mt experimental craft owned by the US Navy, was the largest fully submerged hydrofoil ever built. Propeller driven and powered by 2 LM 1500 gas turbines producing a total of 20.9 Mw, *Plainview* had a cruising foilborne speed of 42 knots, and was designed to operate in sea state 6. Maximum speed was over 50 knots. Maximum hullborne speed, utilizing diesels, was 13.5 knots. When fully loaded the design takeoff speed to become foilborne was 33 knots. Total foil area was 47.4 sq. m.

A fundamental limitation of hydrofoils is imposed by the "square-cube" law. Because the lift developed by a foil is proportional to the foil area (the square of a linear dimension) whereas the vessel weight to be supported is proportional to volume (the cube of a linear dimension), as vessel size increases the foils tend to outgrow the vessel's beam. Cavitation on hydrofoils operating relatively near the ocean surface limits the maximum loading of hydrofoils. Speeds of 50 knots or higher will require development work on advanced foil section shapes.

Another problem with hydrofoils for some naval missions is their markedly greater draft than other hullform types when the vessel is hullborne with the foils down. *Plainview,* for example, has a draft of 7.9 m. Fortunately, its foils can be retracted thereby reducing the draft to just 1.9 m. Total foil system weight for *Plainview,* incuding retracting mechanisms, is 14.9% of the design full load weight. Considerably larger hydrofoil ships have been designed with retractable foil systems, including a US Navy hydrofoil combatant of over 2,000 mt. A hypothetical 1,000-

mt hydrofoil will have a hullborne draft of 10.5 m. Retracting the foils of a large hydrofoil ship is most easily done by rotating the foils up behind the transom and up over the bow. In addition to markedly reducing the draft, retracting the foils makes the propulsion gearboxes and foils accessible for maintenance without drydocking

The HYSWAS (Hydrofoil Small-Waterplane-Area Ship) concept shown in Figure 3.2.1-10 is a hybrid design that combines foil lift with partial buoyant support. Provided that deep draft at slow speeds is acceptable, HYSWAS can provide an attractive combination of excellent seakeeping performance, 40+ knot maximum speed, and substantial range/payload capability. At least two HYSWAS testcraft have been built. In 1994 Kawasaki Heavy Industries built the *Hayate*, a 17.1 m long, one-sixth scale testcraft. Powered by one waterjet driven by a 2835 Kw gas turbine, *Hayate* has a maximum speed of 41 knots. Hullborne draft is 3.1 m. In the U.S., the manned testcraft *Quest* was completed in 1998 and has been tested extensively for the Navy.

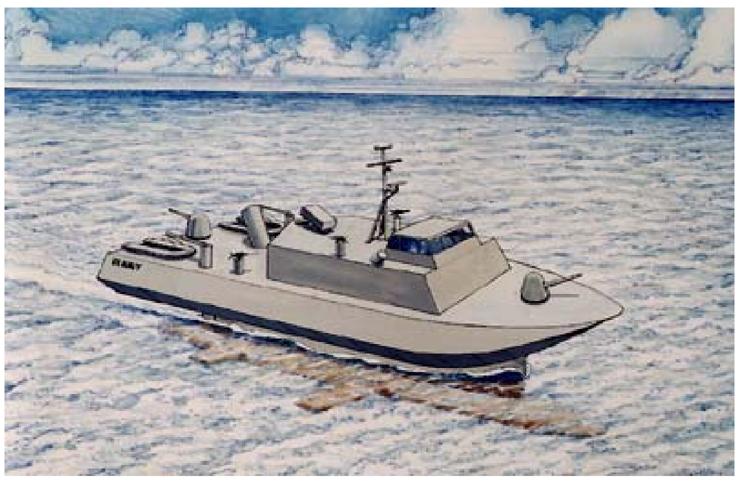

Figure 3.2.1-10: Artist's Impression of a HYSWAS Patrol Boat

Another hybrid design approach is to add foils to multihull vessels, both catamarans and trimarans, as a way of overcoming the limitations on foil geometry imposed by monohull configurations. The wide spacing of a catamaran's hulls allows use of high aspect ratio foils, which are the most efficient. A foil with an aspect ratio of 5 or 6 will have a lift/drag ratio of 14 to 16. In general, designs with a foil-assist fraction of less than 50% are better suited to dual-speed applications than are 80-100% hydrodynamic lift vessels.

Hyundai Heavy Industries relied on 40% foil assist for a 45.5 m long, 35-knot catamaran passenger ferry built in the mid-1990s. Increased hydrodynamic efficiency and seakeeping ability were needed for the specified 700 n. mi. range. In 1999 Halter Marine retrofitted its 45 m long, 175-mt *E-cat* with a pair of foils supporting a fraction of the *E-cat*'s weight, and reported a worthwhile increase in maximum speed. A small Australian builder, North West Bay Ships, delivered the first foil-assisted trimaran, the 54.5 m long ferry *Triumphant*, in 2001 (shown in Figure 2.4.1-1). A pair of foils located near the center of gravity support one-third of the weight

of this 255-mt, 40-knot aluminum vessel. Three waterjets in the center hull transmit the propulsive power from 3 diesels totaling about 7 Mw.

Largest proposed foil-assist installation to date is a retrofit of the 70.4 m long K55 catamaran ferry *Juan Patricio* with two foils: a 16 m span foil with a chord of about 2 m placed near the vessel's center of gravity and a 9 m span foil near the stern. Design full load displacement for *Juan Patricio* is about 650 mt. Total weight of the two fixed foils would be about 13 mt. Estimated performance improvement would be an increase of nearly 7 knots in the vessel's maximum speed, from 48 knots to 55 knots. Total propulsion power is 21.5 Mw.

Surface-piercing hydrofoils are not an acceptable option for a naval vessel that must operate in the open ocean. With a surface-piercing hydrofoil, portions of the foils extend through the air/sea interface when the vessel is operating. Struts connect the foils to the hull of the ship and are sufficiently long to support the hull above the water surface at design speeds. When the vessel encounters a wave, more or less of each foil will be submerged, and the ship will pitch or heave up or down to bring the weight and lift again into balance. These force changes occur automatically, so the surface-piercing hydrofoil system is self-stabilizing. However, foil geometry limits the wave heights in which a surface-piercing hydrofoil can operate safely at high speeds. In addition, motions of a surface-piercing hydrofoil are greater and ride quality worse in moderately rough seas than with a fully submerged foil arrangement.

Planing boats comprise the second type of vessel relying on hydrodynamic lift. A planing hull is designed specifically to achieve relatively high speeds by developing positive hydrodynamic pressures as speed increases, thus generating dynamic lift. When a planing hull is driven at speeds above the displacement speed range the hull initially trims down at the stern. As speed is increased further the hydrodynamic lift increases. Total lift remains equal to the craft's weight because the amount of hydrostatic (buoyant) lift decreases as hydrodynamic lift increases. At fully planing speeds the wavemaking resistance actually decreases as speed increases further. Before reaching full planing behavior there is a characteristic "hump" in the resistance curve of planing boats. This hump is evident in Figure 3.2.1-11.

The volume Froude number scale for the x-axis of Figure 3.2.1-11 is used because planing behavior does not depend upon wavemaking. Nevertheless, the standard Froude number ($V/\sqrt{gL}$) is often used to define the speed boundary for full planing. A ship is considered to be fully planing at speeds above $F_n = 0.90$. At fully planing speeds for a 44.6 mt size boat, Figure 3.2.1-10 indicates lift/drag (L/D) values are between 7 and 7.5. For a 500-mt vessel, the minimum speed for full planing is approximately 55 knots. Consequently, semi-planing hullforms are of more interest for naval ship designs between 500 and 3,000 mt. What is termed the "semi-planing" or "semi-displacement" regime is bounded by Froude numbers between 0.40 and 0.89. Even the 67 m long, 1,070-mt *Destriero* operated in the semi-planing mode during its record-breaking Atlantic crossing at an average speed of 53.1 knots, since the corresponding Froude number is just 0.63.

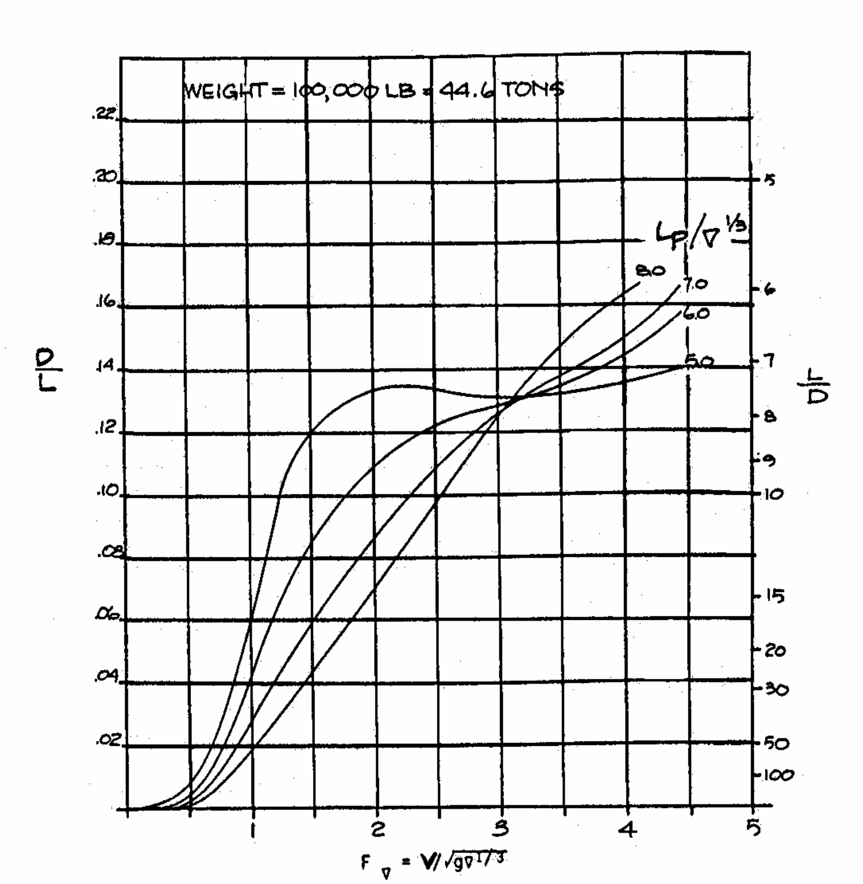

Figure 3.2.1-11: Drag/Lift Contours for Efficient Planing Hulls as a Function of Volume Froude Number and Slenderness Ratio (Savitsky, 1985)

3.2.2 Technology Goals

The objective of the displacement ship and SES powering work is to develop a comprehensive technology base for modern high-speed hulls and validate analytical techniques for prediction of full-scale resistance and powering for HS ship hulls. The following approach will be used:

- Review and analyze existing data to define extensions to analytical models needed for the principal types of HS ship hulls.
- Modify and update analytical models.
- Modify and update test techniques for high-speed hulls.
- Conduct comprehensive model tests to produce data to validate HS ship hullforms and analytical predictions.
- Utilize data from model tests and operational ships to validate predictive techniques by correlation.

Comprehensive tests and analyses will be used to extend analytical methods and validate computer models for slender monohulls ($10<L/\nabla^{1/3}<12$), semi-swath catamarans, trimarans, and high-L/B (L/B of 5 to 6) SES hulls. The approach is to expand current databases and to build upon proven existing analytical models. Potential performance enhancement for SES using mid-cushion transverse seal concepts will be assessed.

Hybrid ship concepts, which most often combine buoyant and hydrodynamic lift, are judged to be a promising area for ships up to about 1,500 mt where further technology development effort is needed. This would encompass model testing, development of analysis and design tools, and development of design standards and practices for both intact and dynamic stability.

3.2.3 Hull/Propulsor Integration

Current practice for designing waterjet-propelled hulls is to first design a hull with low drag followed by the design of waterjets with good propulsive efficiency. Waterjet influence on the hull design is assumed to consist primarily of geometric requirements for the fit of the machinery and inlets. Indeed, geometric fit requirements are important, as is evident in Figure 3.2.3-1, which is a stern photo of the HSS 1500 semi-swath catamaran under construction.

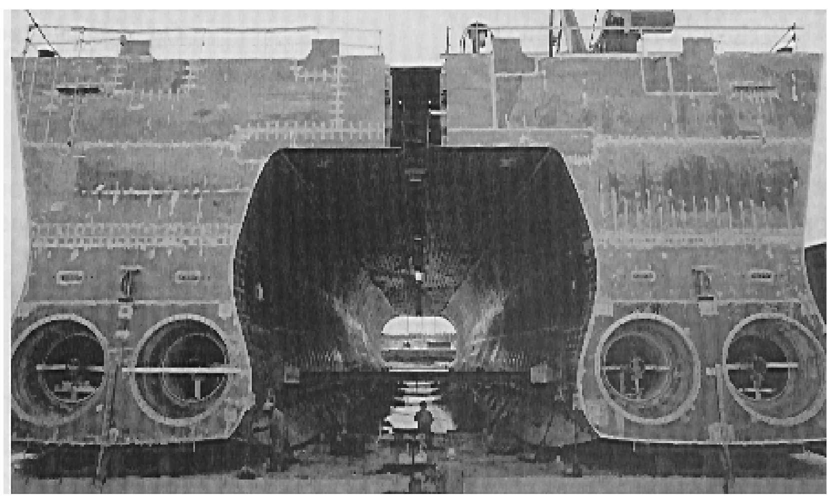

Figure 3.2.3-1: Photograph of the Stern of the HSS 1500 Semi-Swath Catamaran Ferry

For a monohull with a transom stern, operating waterjets have a marked effect on the flow astern of the inlets. Figure 3.2.3-2 shows results of CFD Reynolds Averaged Navier Stokes (RANS) computations of the external streamlines for the flow adjacent to the hull being ingested into twin inlet openings. In addition, the hull flow influences waterjet duct flow and pump entrance conditions. The second illustration of Figure 3.2.3-2 shows the calculated contours of the internal pressure coefficient within the duct of the flush inlet waterjet configuration, simulating operation at the design condition. Flow irregularities in the waterjet inlet are major factors in the design of waterjet components such as inlet ducts, stators, and rotors.

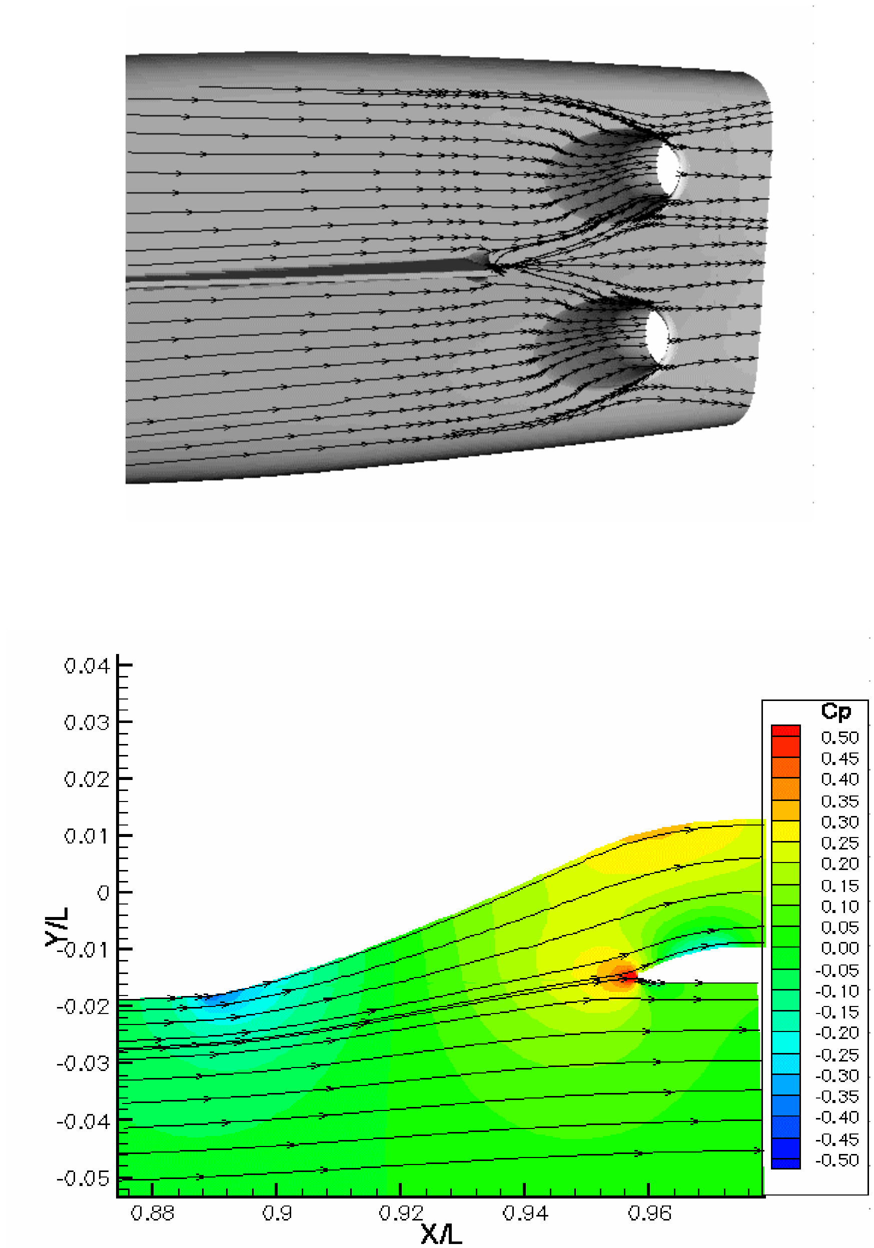

Figure 3.2.3-2: Calculated Effect of Waterjet Inlets on External Flow Streamlines and Internal Coefficient of Pressure

Omitted from the hull design process are the changes in hull flow properties resulting from actual operation of the waterjets. These changes result from alteration of the pressure distribution near the stern caused by waterjet inlet suction under the hull and exhaust behind the transom. Resistance, sinkage, trim, and the direction of the streamlines over the hull are affected. The draw-down of the water surface in the vicinity of the waterjet inlets is of particular concern since it increases the likelihood of air injestion by the waterjets in a seaway. While pertinent to the design of all waterjet-powered designs, the importance of these flow changes is magnified by the slender hulls and high installed power of HS ship concepts. Potential

consequences of this lack of integration include reduced efficiency of the waterjet, higher fuel consumption, and operational limitations in waves.

Hull design should reflect an improved understanding of the actual flow requirements of waterjets. It is common practice to center the waterjet output nozzle at the design waterline. This practice provides the waterjet with a self-priming capability, but also results in deeper transom immersion and increased hull resistance. When the ship is at rest, or just getting underway, self-priming is necessary. However, when the ship operates at moderate to high speeds the flow along the hull bottom ahead of the waterjet inlets is adequate for efficient waterjet operation. Consequently, there is no reason why the ship's control surfaces couldn't be used to raise the transom out of the water, thereby decreasing resistance.

3.2.4 Overview of Development Plan

The technology development effort for displacement hulls will extend the existing technology base to encompass the more slender hulls of high-speed monohulls and trimarans. For catamarans and trimarans the technology developed will include semi-swath hullforms, which have potential for improved seakeeping performance. The needed extensions will be produced using advanced analytic methods, model test data, and available full-scale data. A major objective is to validate analytic models as design tools to support development of slender high-speed configurations. Extension of existing design tools, including Computational Fluid Dynamics (CFD) techniques as well as the model test techniques needed to validate predictions, is a goal of this plan.

Several modern waterjet powered hullforms, including at least one very slender monohull, will be developed using available analytic and empirical data. Different hull concepts will be developed to address key gaps in the existing data base. Resistance, powering, sinkage, trim, and flow data will be measured at model-scale for these hulls to assess resistance characteristics, hull/propulsor interactions, and flow properties such as streamlines on the hulls, flow in the inlets, transom flow, and pressure distribution on the hulls. Comparisons between measured data, estimates produced to develop the modeled hulls, and post-test analysis will be used to establish credibility of the design tools, identify and eliminate shortfalls in the technology, and validate performance of HS ship hulls. Validation will be further enhanced using data for similar hulls developed by commercial projects where available. The following process will be followed in this high-speed displacement hull powering effort:

1. Develop hullform, inlets, and propulsion-system design for alternative HS displacement hulls using analytical methods, model test data, and full-scale data.
2. Prepare model test plans to verify resistance, inlet performance, powering, and performance.
3. Design and fabricate scale models of the hulls and propulsors.
4. Conduct model tests and reduce data.
5. Analyze test data and correlate with performance predictions.

Waterjet inlet simulation tests will be conducted to assess the inlet design, arrangement of the pumps, and hull/propulsor integration. The effects of operation of individual waterjets and combinations of waterjets will be assessed.

The scheduling and costing plan for resistance and powering technology development is shown in Figure 2.2.3-1 for monohulls, Figure 2.3.3-1 for catamarans and Figure 2.4.3-1 for trimarans. The scheduling and costing plan for hull/propulsor integration technology development is shown in Figure 3.2.4-1 for displacement hulls and SES.

The scheduling and costing plan for SES resistance and powering technology development is shown in Figure 2.5.3-1. The SES technology development effort will validate SES analytical design tools for high-L/B HS hulls. L/B for these advanced hulls is 5 to 6. The initial objective is to correlate analytic predictions, model test data, and full-scale trials data for the SES-200, one of the highest L/B SES (L/B ~ 4.2) built to date. Model tests of the SES-200 will be conducted to generate the necessary data. Additional, less meticulous comparisons will also be made using available data for other SES such as the 1,500 mt Japanese *Kibo* (L/B of 3.7), Norwegian MCM (L/B ~4), and commercial lower L/B SES. This will provide the most comprehensive database of model, full-scale and analytical predictions to validate the design tools in the absence of full-scale trials of a high-L/B SES .

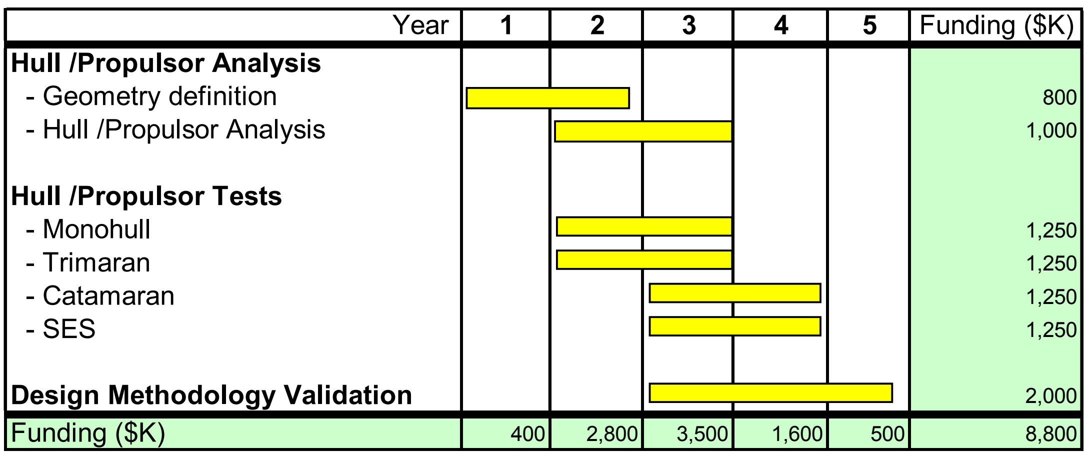

Figure 3.2.4-1: Hull/Propulsor Integration Technology Development Plan

Extension of this technology base to L/B of 5 to 6 SES hulls requires additional model testing. A hullform, propulsion system, lift-air supply system, and seal system will be developed reflecting near-term technology assumptions for the Naval SES. Model tests will be conducted to verify the performance within the design operational parameters. The purpose of these tests will be to verify the integrated performance of the hull, propulsion system, and lift system, and to provide sufficient data for a design database for 2,000 to 3,000 mt high-L/B SES. The following tasks will be performed to support this high-L/B SES effort:

1. Develop hullform, seal concept, propulsion-system design, and lift-system design for a high-L/B HS ship SES using analytical methods, model test data, and full-scale data.
2. Prepare a model test plan to verify powering performance including suitability of transverse seals.
3. Design and fabricate a scale model of the hull, propulsors, and lift system including a method to isolate seal performance from sidehull effects.
4. Conduct model tests and reduce data.
5. Analyze test data and correlate with performance predictions.

A waterjet inlet simulation test will be conducted to assess the inlet design and arrangement of the pumps in each side-hull. The effects of operation of individual waterjets and combinations of waterjets will be assessed.

3.3 Seakeeping

Operational effectiveness of small (500 to 3,000 mt) Naval ships will be affected, and often degraded, by the motions and accelerations they experience at sea. Probably, the largest body of full-scale seakeeping data for high speed ships in this size range has been recorded for various commercial ferries. Most of the commercial ferries have some type of active motion control system. Unfortunately, only a small fraction of this data has been published. The available data pertains to catamarans and deep-Vee monohulls of 700 to 4,000 mt operating at 35 to 40 knots. The demonstrated motion responses and ride quality of these high speed ferries is helpful with regard to burst speed behavior in moderate wave heights. In most cases, the regulatory bodies such as DnV have established limits on the maximum permissible operating speed for a given ferry design as the significant height of the seaway increases. This information provides a useful yardstick for evaluating analytical ship motion predictions.

In reality, small Naval ships will spend relatively little time operating at speeds close to maximum speed. For a Naval ship, seakeeping performance while transiting at 18 to 20 knots in open ocean conditions, and while on station at 1 to 5 knots, is probably more important operationally. Obviously, there is a great deal of experience with the motion behavior of a variety of monohull forms of normal proportions at low and moderate speeds. However, there is little if any data on the motions of very slender monohulls. Hard data on real-world experience with the motion and acceleration characteristics of high-speed catamaran hullforms at low or moderate speeds was also scarce prior to recent trials of leased catamarans by the Navy and Army. With regard to trimarans, the joint UK/USN measurements of motions and accelerations on the 1,200 mt R.V. *Triton* provide one excellent set of 20-knot and low speed data.

The seakeeping behavior and ride quality of low and medium-L/B SES with active ride-control systems installed is known for sizes through 1500 mt. This experience has identified scaling issues that must be resolved to assess seakeeping performance (motions, ride quality) as well as to design the lift system for considerably larger high-L/B SES. Current seakeeping simulation and ride-control system technology has also provided considerable insight regarding SES cushion dynamics. The "bunching" of SES excitation and resonance frequencies in the

seasickness range can lead to a variety of development difficulties. Cushion heave dynamics of a full-scale SES cannot be represented at model-scale due to an inability to scale atmospheric pressure. Consequently, development of design solutions is complicated by an inability to use model-scale motions directly. Current ride-control system analysis and design techniques have demonstrated viable approaches for predicting, evaluating, and controlling cushion dynamics problems up to SES-200 size ships. However, they must be refined and verified for a 3,000 mt high-L/B high speed SES.

3.3.1 State-of-the-Art

The current technology base and capabilities for predicting seakeeping performance will be addressed separately for displacement ships and SES.

3.1.1.1 Displacement Ships

A robust capability for evaluating seakeeping performance of monohull displacement hulls is currently available. Fundamental to this capability are frequency domain computer models based on "strip" theory that assess the statistical properties of ship motions. The SMP95 code was developed in-house by NSWCCD. It predicts 6 degrees of freedom (DOF) absolute motion amplitudes, velocities and accelerations for a monohull ship advancing at constant speed at any heading relative to the waves, in both regular waves and irregular seas. A variety of irregular wave spectra can be chosen. SMP95 predicts relative motions and velocities for specified locations on the ship and it also calculates probabilities of occurrence for propeller emergence or slamming events at specified locations on the hull. Strip theory produces reliable predictions at moderate speeds, but is less accurate at higher speeds.

Three other frequency domain codes available to the Navy are:

- VERES (**VE**ssel **RES**ponse) is a 6-DOF code developed by Marintek in Norway. VERES employs traditional linearized strip theory for ship speeds up to a Froude number of 0.30. At higher ship speeds, corresponding to F_n above 0.40, the user can select a high-speed formulation developed by Faltinsen and Zhao. The high speed theory accounts for interaction with flow from upstream hull sections. Effects of motion control devices can also be evaluated. Unlike SMP95, VERES can make motion predictions for catamarans as well as monohulls. However, low speed motion predictions for catamarans are less reliable because interaction effects between the two hulls are not taken into account.

- The 5 degrees of freedom (DOF) linearized, frequency-domain motion simulation developed by Maritime Dynamics, Inc. can predict motions of an SES hullborne as well as cushionborne. If dummy input variables are used for the lift fan characteristics, this code can predict the motions (except surge) of a pure catamaran. Statistics of vertical plane and lateral plane motions, velocities and accelerations can be calculated at different locations in a specified random seaway. A strength of this code is its ability to quantify motion response reductions from addition of a variety of active or passive motion control devices.

The ship dynamics are linearized about a mean operating trim specified by the user.

- PRECAL (**PRE**ssure **CAL**culations) is a linearized, three-dimensional frequency domain panel code developed by MARIN in the Netherlands for members of the Cooperative Research Group. This code can predict local pressures on the hull as well as motion responses for catamarans (including SWATH-type) as well as monohulls. Capabilities include the effects of passive or active fins on motion responses. A pilot project is underway to extend PRECAL to trimarans.

More complex, time-domain codes are also available for predicting ship motions in a specified irregular wave system. These programs are well validated for conventional monohulls at moderate speeds. A detailed assessment of advanced computational methods for predicting ship motions and loads was published by Beck and Reed (2001). Among the linearized potential flow, time domain codes is SWAN 2 (**S**hip **W**ave **AN**alysis), which was developed at MIT. SWAN 2 is a general purpose, three-dimensional Rankine Panel Method code for the solution of time-domain free surface flows around ships, including high speed monohulls and multihull vessels. Recently, the capability to analyze heave and pitch motion reductions due to adding a passive hydrofoil has been incorporated into SWAN 2. The SWAN 4 code is an extension of SWAN 2 to nonlinear ship motions using the weak scatterer theoretical formulation.

Seakeeping assessments of high-speed displacement catamarans and trimarans are more complex than for monohulls. While the fundamental physics of multihull vertical motions are the same as for monohulls, Navy experience with slow-speed conventional catamarans and SWATH ships has shown that significant extensions to monohull seakeeping technology are required to accurately model features such as between-hull interactions, differences in damping, and different above-water geometry. While prediction capability exists for catamarans, extension of the tools to more accurately model the hull geometry and hydrodynamic effects of semi-swath catamarans and all types of trimarans is needed. Model seakeeping test data at high Froude Numbers is needed for representative HS ship hulls, including semi-swath hullforms, to guide and validate these extensions. Test data at both low and high speeds is also needed for slender monohulls.

3.3.1.2 Surface Effect Ships

Prediction of seakeeping performance of an SES is more complicated than for displacement hulls. Hull design is also a consideration in that side-hull hydrodynamics make a major contribution to ship motions. The early SES test-craft design and development programs identified a number of key aspects of SES cushion system development, including:

1. SES motions differ from those of conventional ships due to the dynamic nature of the air-cushion suspension system, the higher frequencies of encounter, and the catamaran hull-form.
2. SES motions cannot be adequately scaled from tow-tank model test results due to the fact that an important factor in cushion dynamics is the ratio of absolute cushion pressure to

atmospheric pressure. This ratio cannot be properly represented in model tests run at atmospheric pressure.

3. SES cushion systems are subject to a variety of acoustic and structural resonances and instabilities. These effects can limit available gain on an active ride-control system and amplify ship motions.

Since control of ambient air pressure and density in large tow-tank facilities is not economically feasible, SES development programs have adopted the concept of computer program simulation for all parameters that are affected by cushion compressibility. The motion simulation programs are applied to predict results of model tests and existing test-craft operations. If good agreement is achieved, the simulation programs are considered valid for prediction of cushion pressure variations and motions at full-scale.

Analysis and design efforts under the early test-craft programs recognized the unusual nature of SES motions and the need for active control systems to improve ride quality and habitability. The effect of SES motions on crew performance was quantified by applying simulated full-scale motions (with and without ride-control) to volunteer subjects for extended periods. These tests confirmed the need for SES ride-control to reduce crew fatigue and motion sickness. Cushion resonance and instability problems resulted in severe gain limitations and reduced effectiveness for the initial XR-1, SES-100A and SES-100B ride-control systems.

A digital microprocessor-based ride control system, driving deck-mounted vent valves, was installed on the SES-200 and an extensive test and evaluation program was conducted during 1983. Ship motions and ride quality data were acquired under a wide variety of operating conditions. The effectiveness demonstrated by the SES-200 ride-control system during these tests is illustrated by the fact that r.m.s. cushion pressure variations were reduced by up to 60% and r.m.s. heave accelerations at the C.G. were reduced by nearly 50% under some conditions.

The largest SES built to date, the TSL-A at 1,500 mt, uses a combination RCS involving both vent valves and T-foils.

The primary design tools applicable to the SES motions/ride quality/cushion dynamics area consist of digital computer programs for simulation of SES dynamics. These programs typically accept input which defines an SES of any size and hull configuration. In addition, input parameters, or "option" modules, allow selection of alternate lift systems, bow and stern seals, and ride-control systems. The Navy has a 5 degrees of freedom (DOF) linearized, frequency-domain SES motion simulation developed by Maritime Dynamics, Inc. under contract. The program predicts the statistics of vertical plane and lateral plane motions (no surge) for an SES operating in a random seaway with or without ride-control. The ship dynamics are linearized about mean operating conditions for each case (trim, draft, cushion fan flow, etc.) that are established by operator inputs.

Other programs are currently available, including:

- SES 5-DOF Seakeeping Program (Maritime Dynamics, Inc.) – This design tool is a linearized, frequency-domain SES motion simulation. SES Finite-Volume Vertical Plane Motions Program (Maritime Dynamics, Inc.) – This program was developed to

predict the effects of cushion acoustic and hull modes on active ride-control systems. The program represents cushion dynamics by a one-dimensional finite-volume model. Effects of lower frequency longitudinal acoustic modes as well as flow lags in fan and vent ducts are included. The non-linear character of certain phenomena is retained, including side-hull leakage flow and vent-valve position limiting. Active ride-control is simulated as independently-controlled vent valves located at several longitudinal cushion locations.

- Time-Domain SES Heave Dynamics Program (Band, Lavis & Associates) – As part of the 1987-1989 SES Hullform Technology Program, BLA developed a time-domain representation of the heave dynamics of air-cushion supported craft. This representation allowed the effects of scale, compressibility, associated water mass and side-hull hydrodynamics to be studied individually. It demonstrated once more that large SES are very much more prone to heave instability than SES of 200 tons or less, and that active ride-control will likely be required for large SES.

- 6-DOF SES Motion Simulations (Oceanics, Inc.) – The "6-DOF" program developed by Aerojet and Oceanics, and used in development of the RMI 3KSES, is a non-linear, time-domain simulation of SES motion. The program predicts all rigid-body motions as functions of time from the starting condition. The analytical representations for hull hydrodynamics, bow and stern seals, lift fans, propulsors, and ride-control components are non-linear and considerably more complex than those used in the frequency-domain programs.

- 6-DOF SES Motion Simulations (Textron Marine) – The Textron Marine program is similar to that of Oceanics, except that bow and stern seal dynamics are based on empirical data rather than analytical modeling. Given a realistic starting condition, this program is particularly useful for tracking ship motions during discrete wave encounters and slamming and broaching events.

The existing SES computer simulation tools will require some further development to be applicable to a 3,000 mt high-L/B SES, but should be capable of providing valuable insights into the effects of scale, which can more easily be accommodated in computer simulation than in the model test tank. The fully-developed programs will be applicable for predicting seakeeping and ride quality of a combatant SES, and for evaluating cushion pressure variations, both statistically and that resulting from discrete wave encounters. Issues related to cushion dynamics and control have previously been addressed by Aerojet, Bell, and RMI for the 2K/3KSES, with extensive simulation support. Notable examples include:

- stern-seal dynamics and flutter
- waterjet air ingestion in waves
- lift-fan drive-train dynamics as driven by wave pumping and fan flow control

Development of routines addressing these effects is needed for insertion into motion prediction programs.

To support an integrated approach to SES cushion dynamics, development of "modular" program(s) is needed which will include simulation of basic SES cushion dynamics and motions as a framework for related subsystem dynamics evaluation. Program options will allow emphasis on selected subsystem phenomena and a reasonable level of input complexity when applied to specific dynamic problems such as those noted. The simulations will also be capable of evaluating potential corrections and improvements via the control of cushion dynamics.

3.3.2 Technology Goals

Wave-induced motions of high-speed displacement hulls and high-L/B SES hulls will differ from those of conventional ships. Extensions to the technologies used to predict ship motions for these hullforms are needed, including sinkage and trim effects, to develop ship designs that can satisfy mission requirements in representative sea conditions. These predictions are also needed to design essential subsystems such as motion-control systems, SES lift fans, and SES bow and stern seals.

A modest extension of displacement monohull seakeeping technology is needed to validate the accuracy of existing seakeeping predictions for hulls with the greater slenderness and larger transoms of HS ship hulls.

A more significant extension of seakeeping technology is needed for proposed Naval catamarans and trimarans. Extensions are needed to reflect the geometric, hydrodynamic, and mass property characteristics of modern multihulls with a variety of hull section shapes, including the semi-swath type. Analyses must include trim and sinkage effects due to high forward speed as well as the beneficial effects of different motion control devices.

The SES seakeeping technology development goals are to extend the capabilities of existing motions prediction computer programs to address issues critical to a 3,000 mt high-L/B, high speed SES design including cushion resonance/instability predictions, cushion/water interface dynamics, and modular simulations for subsystem dynamics. These technology extensions will allow integrated development of SES hulls, lift systems, seals, and ride-control systems with the seakeeping performance required for HS ship missions.

3.3.3 Overview of Development Plan

Displacement hull seakeeping technology development includes extension of analytical tools to address the higher slenderness and large transoms of high speed monohulls as well as extensionsof other tools to model the hull geometry and hydrodynamic effects of semi-swath catamarans and all types of trimarans. The software developed will be validated with existing model test data as well as test data developed as part of this plan. Model seakeeping test data at high Froude Numbers is needed for representative HS ship hulls, including semi-swath hullforms, to guide and validate these extensions. Test data at both low and high speeds is also needed for slender monohulls.

The scheduling and costing plans for displacement hull seakeeping technology development are shown in Figure 2.2.3-1 for monohulls, Figure 2.3.3-1 for catamarans and Figure 2.4.3-1 for trimarans.

SES seakeeping technology development includes development of cushion resonance/instability predictions, cushion/water interface dynamics predictions, and modular simulations for subsystem dynamics. Analytical predictions of motions and ride-control performance will be compared with model test data and full-scale experience. The overall program is broken into three phases. The first phase will structure the problem and will develop the basic analytical approaches and routines that can be used to evaluate the feasibility and significance of the expected result. The second phase will incorporate the analysis into one of the existing SES motion simulations. Simulation results will be correlated with available test data. Requirements for experimental validation testing will be established. The third phase will provide experimental validation and demonstration of the simulation predictions.

The scheduling and costing plan for SES seakeeping technology development is shown in Figure 2.5.3-1.

3.4 Maneuvering

High speed displacement hulls and SES concepts differ from existing ships in several factors that affect maneuvering, dynamic stability and control. Use of more slender hulls, use of waterjets with nozzle control for maneuvering, and selective use of small rudders for high-speed maneuvering all influence initial stability, dynamic stability in waves, and dynamic stability in turns. Other factors affecting stability in waves and in turns include high forward speed, the effects of transverse CG shift, roll inertia, and turn rate. The objective of maneuvering and dynamic stability technology development is to provide assurance that Naval high-speed ships can operate safely throughout their operational envelopes.

3.4.1 State-of-the-Art

Technology to predict the maneuvering and dynamic stability characteristics of displacement hulls at low and moderate speeds is well established. The approach used requires solution of generic equations of motion formulated with empirically or experimentally-derived hydrodynamic coefficients. The resulting system of equations can then be analyzed to assess conformity with U.S. Coast Guard, Code of Federal Regulations, IMO, and classification society requirements. While the equations of motion are general, the hydrodynamic coefficients are hullform specific. In a typical waterjet propulsion system installation there are no appendages such as shaft brackets and rudders near the stern of the hull. Absence of these appendages decreases the ship's lateral stability.

3.4.1.1 Displacement Ships

Existing regulatory body requirements were developed for slower, less slender monohulls. Structural requirements and crew ride quality considerations often result in these conventional ships reducing speed in higher seas. This combination of current standards and operating

practices results in assurance of adequate maneuvering and control authority to assure safe operations for the ship loading conditions, speeds, and sea conditions encountered.

However, the higher speed, greater slenderness, draft variations, and control systems envisioned for high-speed Naval displacement hulls may result in unstable dynamic behavior modes that do not occur for the more conventionally-designed and operated hulls. Furthermore, the premium attached to burst speed will encourage maintaining high speeds in high seas. In 1997 the Italian ferry builder Fincantieri undertook a theoretical and experimental investigation of the coursekeeping ability of fast deep-V monohulls powered by waterjets (Brizzolara, 1998). Towed PMM (Planar Motion Mechanism) tests were carried out on a model of the Fincantieri MDV 1200 design without waterjets or waterjet inlets. Lateral force, yawing moment and yaw rates were first measured for the bare hull. Then, data was obtained with twin anti-roll fins appended aft of amidship and, alternatively, with twin fixed vertical fins at the transom. A coursekeeping simulation was developed which added theoretical formulations for the influence of waterjet inlets and waterjet steering forces to the derivatives of the hydrodynamic forces measured in the PMM experiments. It was determined that the contribution of waterjet inlets to course stability is of the same order as that resulting from adding a small vertical fixed fin at the transom. The waterjet inlet "fin effect" is due to the change in momentum of water entering the inlet with a certain transverse velocity component and exiting with only a longitudinal velocity component. A second finding was that placing anti-roll fins at the proper location aft of amidship is important for acceptable coursekeeping. The paper by Brizzolara also presented limited full-scale data from spiral maneuvers of the MDV 1200.

Systematic evaluation of significantly more slender, high-speed monohulls to assess the possible existence of undesirable stability characteristics in calm water and in waves has not been done. Ensuring that a very slender monohull has adequate intact roll stability in waves may require the addition of small buoyant outriggers. This approach was taken for the high-speed sealift design developed by NSWCCD and shown below. The pair of side hulls in this design contribute only 2% of the total ship displaced volume.

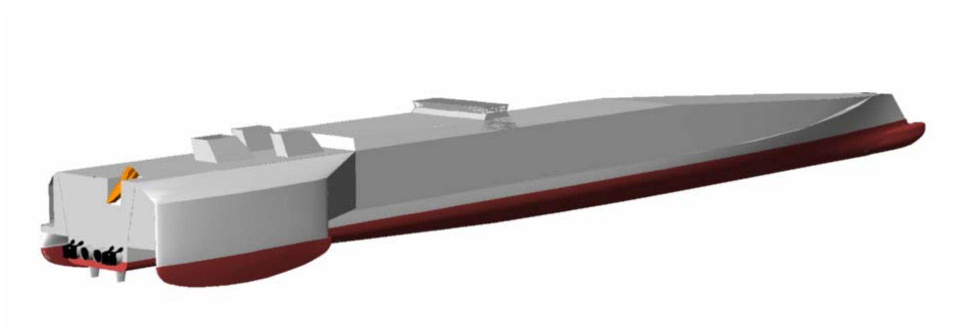

Figure 3.4.1-1: NSWCCD High-Speed Sealift Ship Design

For a trimaran with larger side hulls, turning ability is the key concern, rather than stability or coursekeeping. This is because the two narrow side hulls of a trimaran act like two fixed fins when the trimaran maneuvers. Their presence has a positive effect on coursekeeping and a detrimental effect on turning ability. Results of a pioneering investigation of trimaran maneuvering was published by researchers at University College London (Zhang and Andrews, 1998). They used a combination of theoretical and experimental methods to quantify the turning ability of a notional 5400 mt trimaran at 20 and 30 knots. Model tests were carried out for two longitudinal locations of the side hulls and two different side hull drafts. Moving the side hulls further aft, or increasing their submergence, was found to increase the trimaran's turning radius somewhat. Use of differential propulsion in the side hulls was found to be very efficient in turning the ship at very low speeds.

While there is a wealth of full-scale experience with high-speed catamaran ferries, relatively little has been published. Full-scale maneuvering trials of a 45 m long catamaran ferry were carried out about 4 years ago near Lisbon, Portugal (Soares, 1999). These trials were part of a European research program. In 1993 Japanese researchers publisher a paper containing rotating arm model test data and maneuvering simulation results for a "super-slender, twin-hull" ship design (Ishiguro, 1993). The simulation results were compared with full-scale measurements for a 30 m long prototype.

Existing Navy displacement hull maneuvering tools are monohull-based and lack the capability to model catamaran or trimaran geometry and mass properties. Consequently, while the methodology to analyze maneuvering and control of high speed displacement ships exists, tool extensions to encompass multihulls, additional measured hydrodynamic coefficient data, and analyses are needed to assure safe operations of HS ship concepts throughout the operating envelope.

3.4.1.2 Surface Effect Ships

The capability to predict the dynamic stability and maneuvering characteristics of SES via a combination of model testing and computer simulation has reached a high level of maturity. A wealth of test experience has been accumulated over the past forty years, but, until recently, this has mostly been limited to the characterization of specific designs with little attempt or opportunity to explore systematically any wide variation in hullform or basic stability parameters.

In recent years, advances in computer-aided analysis have permitted more extensive procedures to be developed for treating the non-linear behavior of the SES. With such tools and testing techniques, SES can be designed to exhibit adequate static and dynamic stability in both the intact and damage condition while both cushionborne and hullborne. When hullborne, this is due to the large initial waterplane moment of inertia provided by the wide separation of the side-hulls and the relatively small clearance of the wet-deck, which results in the cross-structure entering the water after only a few degrees of list. The resulting increased waterplane limits the impact of off-center flooding and sinkage; consequently, larger subdivision lengths are acceptable on SES designs than on equivalent-sized monohulls.

As SES become larger, for moderate design speeds, the preferred length-to-beam ratio tends to increase on account of the advantages gained in the form of reduced resistance. High cushion heights are also desirable for large ocean-going SES to keep the wet-deck clear of large waves. High wet-deck heights tend to imply high vertical CGs. The combined effect has been to develop high, narrow ships for which on-cushion dynamic roll stability during turns and in synchronous beam seas, especially in adverse weather, has become of greater concern.

In recognition of this trend, recent large SES designs have generally featured side-hulls of relatively larger volume to increase stability and, in addition, they have been able to accommodate heavy machinery relatively low within these side-hulls to lower the center of gravity. For the range of small SES built to date, the side-hulls have generally been too small for the installation of much machinery, which must instead be located above the level of the cross-structure wet-deck, which has resulted in a relatively higher vertical center of gravity (VCG). In addition, for large SES, all the fuel is located in the lower extremities of the side-hulls to help lower the VCG in the full fuel-load condition.

Important features affecting dynamic stability also include the side-hull length, volume and deadrise, the types of bow and stern seals, the size and location of skegs, fences and rudders (if included), the type of propulsion system, the type of maneuvering system, and the ship's moments of inertia.

The primary circumstances leading to a risk of unfavorable dynamic behavior include high-speed turning maneuvers, sudden helm reversal and/or sudden propulsor or steering-system failures at high speed, running with high winds and synchronous seas on the beam, and operation in very steep following or quartering seas.

Research conducted in the UK has generated a greatly improved understanding of overall on-cushion stability requirements to the extent that provisional criteria based on practical and purely numerical methods have been set. Considerable progress has been made towards a comprehensive understanding of the dynamic stability of SES. However, areas that have been identified that would benefit from further examination during the design-development phase. are:

- Development of improved air flow leakage models, especially over the first 10 degrees of heel, to improve maneuvering and control simulations.
- Stability in waves when large roll angles cause substantial transverse shifts in payload.
- Stability in turns as affected by ship speed, the turn diameter, and bow-up trim.

3.4.2 Technology Goals

Maneuvering and control characteristics of the large slender displacement hulls and high-L/B SES hulls needed for HS Naval missions will differ from existing ships. Extensions to the technologies used to predict maneuvering and control performance for these ships are needed to develop hull designs that will safely and efficiently transport crew and cargo at high speeds in

representative sea conditions. These predictions are also needed to design essential subsystems such as steerable waterjets and high-speed steering rudders.

Modification of existing displacement hull maneuvering analysis tools is required to model multihull geometry and mass properties and include the effects of large steerable waterjets. Modest extension of maneuvering technology is needed for monohull and multihull displacement hulls to generate the hydrodynamic data needed to make predictions for the slender, high-speed hulls. Analysis and testing of HS hulls are required to identify and eliminate undesirable modes of transient response in calm water and in high seas.

SES maneuvering technology development goals are to provide the capability to develop SES for HS Naval missions with adequate maneuverability and controllability in all modes of operation (on-cushion, off-cushion and partial-cushion) and applicable sea states. Issues to be resolved include the amount of required steering, reversing capability, and performance in quartering seas.

3.4.3 Overview of Development Plan

Displacement hull maneuvering technology development includes extension of monohull tools to model hull geometry and mass properties of multihulls as well as generation of hydrodynamic data for the slender, high-speed hulls. The effort consists of software development that incorporates existing model test and full-scale data as well as test data developed as part of this plan. Scheduling and costing plans for displacement hull maneuvering technology development are shown in Figure 2.2.3-1 for monohulls, Figure 2.3.3-1 for catamarans and Figure 2.4.3-1 for trimarans.

SES maneuvering technology development consists of extending existing SES capabilities through a program of design, analysis and model testing. Major issues to be resolved are the effects of cushion airflow on initial stability, the effects of transverse CG shift, roll inertia, and forward speed on stability in waves, and the effects of forward speed and rate of turn on stability in turns. Data from existing craft, including the SES-200, Techno Superliner-A and others, will be used to validate analytical simulations at larger than model size.

The scheduling and costing plan for SES Maneuvering is shown in Figure 2.5.3-1.

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4.0 LOADS, MATERIALS, AND HIGH-STRENGTH/LIGHTWEIGHT STRUCTURES

4.1 Introduction

The operational requirements for proposed small high-speed ships dictate non-conventional hull forms with very low structural weight fractions that can only be obtained using innovative structural configurations, materials and manufacturing processes. Unfortunately, because of the lack of experience with many of the candidate hull forms, structural configurations and materials, structural loads and response are unknown. An optimum reliable structural design can, therefore, not be achieved. A coordinated research and development program is necessary to determine loads and develop structural technologies for near and mid-term, small high-speed naval ships that meet these goals and ensure structural integrity.

To design a marine vehicle, one must know the seaway and deck loads, as well as the structural response and the properties of the structural materials. Although a great deal is known about the seaway loads on conventional ships (Figure 4.1-1) operating at slow to moderate speeds, the effects of 40-knot speeds are less well understood. Effects of 50 or 60-knot speeds are largely unknown. Only a small database exists for seaway loads on novel hullforms such as trimarans.

Figure 4.1-1: Seaway Loads

To attempt to design a ship outside of our current experience base, that is, operating at very high speeds or with a novel hullform, without model tests can lead to two unacceptable results: (1) under-predicting the loads so that the ship suffers significant (and possibly catastrophic) structural failure, and (2) applying excessive factors of safety (to cover ignorance levels) leading to an overly heavy structure.

The materials and structure required to resist the seaway loads have historically been dictated by cost, weight, and producibility considerations. Moderate to large size conventional ships are built of steel, a relatively low-cost but high-weight material. Smaller, weight-critical craft have been built out of aluminum or fiber-reinforced plastic (FRP) composites since the 1950s. The lightweight materials can save weight, which can be used to increase the ship's speed, range or payload. Since the structural weight of a ship is 20% to 40% of its displacement, the potential payoffs in weight savings are substantial – in the hundreds of tons.

To realize these weight savings, a significant research and development effort is necessary to resolve a number of issues related to strength, fatigue and fire resistance. Projected weight savings and corresponding deadweight density increases are shown in Figure 4.1-2 for near-term small high-speed Naval ships and are compared with that of existing ships and aluminum ferries.

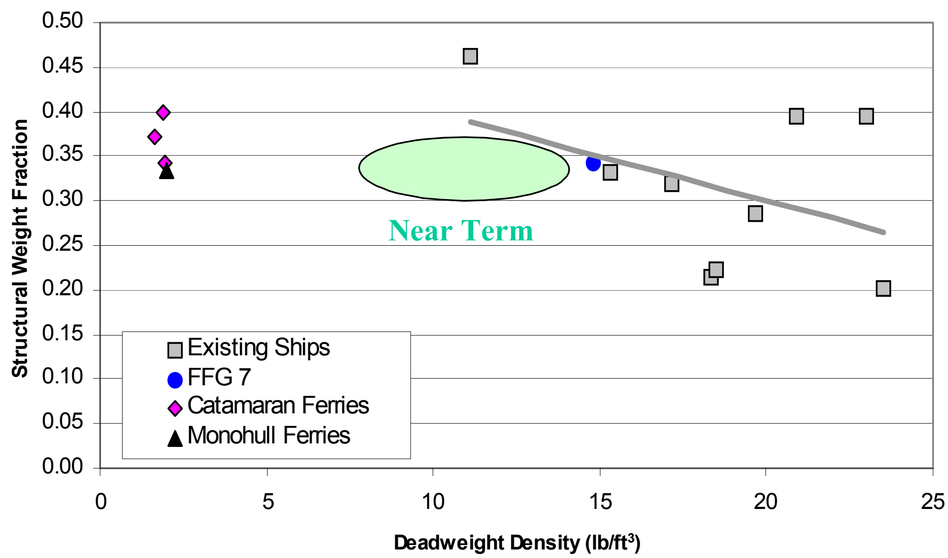

Figure 4.1-2: Structural Weight Fraction versus Deadweight Density

Numerous research and development issues related to loads, materials, and high-strength/ lightweight structures need to be addressed to realize these weight savings. These issues are outlined in Figure 4.1-3 with approximate effort levels identified. The individual tasks will be described in greater detail later in this section.

Ultimately, design procedures for novel hull forms that include innovative structural technology and material options must be formulated to ensure reliable, predictable performance. It is probable that the material characteristics, loads and structural response for a selected design may exhibit more variability than current designs. Consistent design guidelines and procedures must be developed to ensure comparable reliability levels compared to current designs.

Year	1	2	3	4	5	Est. Cost ($K)
Develop structural concepts						600
Identify new/emerging materials						400
Loads Requirements & Design Criteria						
- Loads testing *(cost included with hull form technology)*						0
- Conditioned-based monitoring and inspection						3,000
-Develop design guide						800
Manufacturing and Material development						
- Develop reliable, strong, lightweight materials						2,000
- Develop & optimize material ruggedness characteristics						1,000
- Demonstrate manufacturing processes & approaches						8,000
- Determine & optimize structural & fatigue performance						3,000
- Improve joining technology						3,000
- Fire performance						2,000
Funding ($K)	1,800	6,100	9,500	5,400	1,000	23,800

Figure 4.1-3: High-Strength/Lightweight Structures Technology Development Plan

4.2 Seaway Loads

There are two kinds of seaway loads acting on ships: primary and secondary. Primary loads are bending and torsional moments which flex and twist the hull as if it were a beam or girder. The interaction of the wave buoyancy forces and the weight of the ship cause bending in the vertical plane (hogging and sagging); see Figure 4.2-1. Bending in the transverse plane (lateral bending) and torsional twisting is caused by port to starboard differential buoyancy and rolling in oblique seas. Transverse plane loads are particularly important for multihulls and SES.

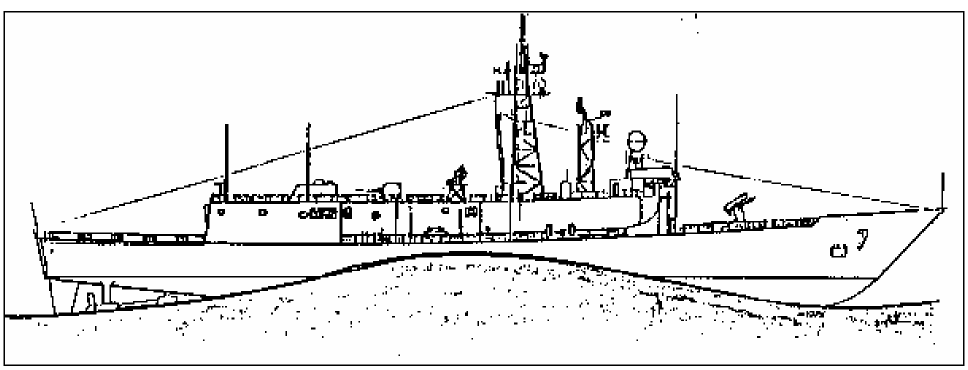

Figure 4.2-1: Hogging Bending Moment

Historically, conventional monohulls have been designed by developing shear and bending moment diagrams from a static balance of the hull girder on a standard wave. Damage during sea trials of the CVA 9 *Essex* in the late 1950s led to a series of full-scale trials and model tests to define a dynamic component (whipping) from slam impacts that increase the vertical and

lateral bending moments along the length of the ship. The slam-induced whipping is exacerbated by speed and, in some cases, can approach the magnitude of the wave-induced moments. All of the primary hull girder moments increase in proportion to the square of the length of the ship. Longer, more slender hulls that reduce wavemaking resistance will experience higher primary wave loads.

Secondary loads consist of the static and dynamic pressures acting on local structure. Hydrostatic pressures are caused by the head of water from hull submergence and passing waves, and are functions of ship draft and sea state. At slam impacts, a hydrodynamic pressure is caused by large bow motions (pressures then act on the bottom of the bow as it re-enters the sea, when the bow flare is immersed, or when multihull/SES cross-structures are immersed). Wave slapping pressures can be significant on the hull sides and transom. Green sea loadings occur when waves crash over the bow, striking the weather deck and front of the deckhouse (see Figure 4.2-2). All of these hydrodynamic pressures are functions of hull geometry and increase with ship speed and sea state. Several of the large, high-speed ferries have experienced structural damage in service due to slam loads. Two examples are the Corsaire 13000 monohull (Figure 2.2.1-2) and one of the large semi-swath catamarans (Figure 2.3.1-2).

Figure 4.2-2: Green Sea Loading

4.2.1 State-of-the-Art

4.2.1.1 Displacement Ships

There are a number of analytical tools available for predicting seaway loads for conventional monohulls. SMP95 is a linear strip theory code in the frequency domain that gives good results for the vertical wave-induced portion of hull girder bending, but lateral bending moment results are suspect and the code is not applicable for whipping effects. Other frequency domain codes that are available are VERES and PRECAL. Although VERES is similar in some ways to SMP, it includes a high speed formulation and a nonlinear version is available. While PRECAL is

representative of the state of the art in 3-D frequency domain codes, the U.S. Navy has limited experience with its use.

With regard to time domain codes, a representative sample includes QLSLAM, DYNRES, LAMP, and THAFTS. All of these codes can include the effects of slam induced whipping of the hull girder bending, but have some limitations. Specifically, QSLAM is a quasi-nonlinear code and DYNRES is limited to 2-D. LAMP and THAFTS are 3-D codes which require a fair amount of expertise and can be time consuming to run. Of these two codes, NSWCCD has had experience only with the LAMP code. CMSLAM and SLAM-2D can be used to analyze hydrodynamic impact problems. Navy experience with both codes is very limited. Either code can be used to solve both symmetric and asymmetric impact problems, but CMSLAM has been found to run significantly faster in solving symmetric loading cases.

All of the analytical codes mentioned were originally developed for conventional monohulls. Some codes have limited validation while others have been extensively validated. The newer time domain codes have had the least validation. Although most of these codes can be used to analyze multihull forms, additional enhancements and validation efforts are necessary. Additional extensions to the technology are required to model geometry and mass properties of high speed trimarans and catamarans. Model test data is required for small HS displacement hull concepts to guide development of the analytic models and validate the predictions. They need further validation (and possibly modification) for applications to novel hullforms.

Model tests can be used to predict primary and secondary loads for conventional and novel hullforms under extreme sea and operational conditions. The test data are analyzed and presented in a probabilistic format that can account for such variables as expected lifetime, sea conditions, and operational parameters.

4.2.1.2 Surface Effect Ships

A 3,000 mt SES would be somewhat smaller than the 140 m long, 38-knot SES for which Mitsui Engineering & Shipbuilding was awarded a building contract in January, 2003. This 4,000 mt ship is scheduled to begin service in Japan in 2005. Currently, the largest SES in operation is the 1500 mt, 74 m long *Hisho*/*Kibo,* also built by Mitsui. As with displacement hullforms, higher design speed will increase the magnitude and frequency of slam loads compared to a conventional ship. In addition, the traditional spread between hull primary response frequency and the wave encounter frequency will narrow. These factors may result in slamming becoming a major contributor to the determination of design primary bending moment.

While slamming will contribute to primary design bending moments, it will contribute in a different way than it does for more traditional hullforms. The bow seal and the cushion protect the SES hull from slams at speed. However, operation without the bow seal or the cushion at low speed will still result in slam loads that must be considered in the primary design bending moment.

Although most full-scale SES experience is limited to hulls of considerably smaller size than the proposed HS combatants, a large body of experimental, analytical and design data has been

developed for large SES concepts. In the U.S., comprehensive development of the SES concept began with the SES-100A and SES-100B programs in 1965. As these programs evolved, a number of different approaches were explored to develop hydrodynamic design loads for SES. Initially, because there was no historical experience, a linear, frequency-domain seakeeping model was used to predict the frequency of bow and wet-deck slams and the associated relative velocity at impact. This information was used as input to a 6 degree of freedom, time-domain impact model that used an adaptation of seaplane theory to predict pressure distributions, total loads, motions and accelerations. Limited full-scale trials of the SES-100A and SES-100B were used to validate predicted values. Later, during the 2KSES program, bending moments and stresses were measured during segmented and "grillage" model tests, and a full-scale section of the wet-deck ramp was tested to determine the stresses that would be experienced during high-speed impacts. The "grillage" model was built to model the structural elastic characteristics of the full-scale RMI 2KSES and was tested in the on-cushion and off-cushion models of operation at all headings to the waves. These tests showed that few wet-deck impacts were experienced in the on-cushion condition. However, the loads and bending moments experienced from wet-deck impacts in the low-speed off-cushion condition were found to be considerably higher than the high-speed on-cushion loads.

4.2.2 Technology Goals

As hullforms are introduced outside of our current experience base, model tests will be required to determine the proper loads and response. These model tests will expand the experience base, enhance our analytical capabilities, and lead to a reliable, efficient hull structure. The U.S. Navy is currently evaluating the performance of high speed ferry catamaran designs for military applications. The response of the structure is being monitored as part of the evaluation. Ideally, the loads on the structure will be determined for appropriate operational scenarios, enabling a more efficient, reliable structural design in the future.

Although model tests are the key to understand the loads and design an optimum structure to resist the loads, a number of load reduction strategies to reduce the primary and secondary seaway loads are also being considered to reduce the hull structural weight. For example, hull or bow forms that reduce slamming and/or primary loads can save structural weight as well as improve seakeeping and resistance. Often slamming events and the load increase associated with those events occur only in certain sea conditions and ship directions. If those conditions can be avoided, extreme and design loads can be reduced, leading to a reduction in structural weight. Preliminary indications from measurements aboard some of the aluminum, high speed catamaran designs indicate that slamming occurs in specific sea conditions and ship directions. Trade-off studies are needed to determine the structural weight reductions achievable if operational limitations are introduced to avoid these conditions. A reliability based format should be developed to perform such assessments.

Condition-based monitoring methods are also being utilized and developed to reduce the structural weight. Implementing real-time wave measurement systems to avoid damaging sea conditions can reduce primary and secondary loads. Active systems are being investigated to reduce the whipping component of the hull girder bending moment. In addition, the design allowable stresses can be relaxed if frequent, focused inspection schedules are conducted,

automated hull inspection and repair systems and techniques are implemented, and strain gauges and sensors are used to monitor the hull structural behavior during its operation. Further research and development in these areas is needed to realize the potential weight savings and the impact on the structural reliability.

4.2.3 Development Plan for Loads

The structural loads and response of novel hullforms at very high speeds are unknown and need to be determined to avoid over-design or catastrophic failures. The first phase of determining the loads on these novel hullforms is beginning with the evaluation of the high speed aluminum catamaran vessels currently in service. Because of the need for a very low structural weight fraction, optimal structural performance is required for high-speed Naval missions, making the determination of these unknown loads very important.

The increased understanding of the structural loads and response leads to the development of design guidelines, the investigation and implementation of load reduction strategies, and the development and adoption of active, strain monitoring systems and focused inspection schedules to provide a reliable and optimum lightweight structure.

The effort needed to develop design guidelines, determine structural loads and response, and develop load reduction strategies and monitoring procedures and systems to ensure a reliable, lightweight structure is shown in Figure 4.1-3. This plan includes:

- Investigation of existing commercial, high speed ferry designs to determine loads and response.
- Model tests of various hull stiffnesses and geometries, speeds and headings to determine primary and secondary loads for high-speed operations and novel hullforms.
- Analytical code verification and modification.
- Investigation and development of active systems to reduce the whipping component of the hull girder bending moments.
- Development of wave measurement systems and load avoidance and monitoring strategies that will reduce hull girder loads and, therefore, structural weight.
- Investigation of structural modifications such as an articulated hullform to reduce the primary loads.
- Development of automated hull inspection and repair systems, implementation of focused inspection schedules, and the analysis of their effect on structural weight and reliability.
- Development of design guidelines, using the results of the various investigations, optimizing structural reliability and minimizing weight.

4.3 Materials

4.3.1 State-of-the-Art

Large conventional monohull ships are predominantly constructed of steel, while smaller weight-critical vessels (under 130 meters) are frequently constructed of aluminum or composites. Most of the smaller SES constructed to date have used aluminum as their basic structural material. Exceptions include the Vosper Hovermarine SES (the HM-2 and HM-5 series) that are constructed principally of glass-reinforced plastic. Weight-critical ships frequently use aluminum material to reduce weight because it has one-third the density and modulus of steel and a fatigue allowable stress one-half that of steel. As a first level approximation, for ships governed by hull girder bending (ships over 130 meters long), aluminum can save one-third of the structural weight of a steel vessel. There are no technical reasons why large ships cannot be fabricated from aluminum, but consideration must be given to the relatively low fatigue characteristics of aluminum, the large deflections that aluminum structures exhibit compared to steel structures and the cost of aluminum, which is five to eight times more expensive than steel. The 140 m long, 4,000 mt Japanese SES scheduled to begin service in 2005 will be built of aluminum.

The U.S. Navy has been reluctant to use aluminum construction on ships intended to go into harm's way since the 1975 collision and resulting fire aboard the USS *Belknap* (CG 26) following it's collision with the *John F. Kennedy* (CV 67) (see Figure 4.3.1-1). In addition to fire concerns, the U.S. Navy has also documented a significant number of fatigue problems associated with the use of aluminum in combatant superstructures that has added to its reluctance to use aluminum despite the potential weight savings offered by the material.

Figure 4.3.1-1: USS *Belknap* (CG 26) after colliding with the *John F. Kennedy* (CV 67)

However, the successful use of aluminum as the primary structural material at unprecedented lengths for aluminum ships (e.g., the 125 m long Alhambra II monohull fast ferry and the 126 m lomg HSS 1500 Semi-Swath catamaran) is causing the U.S. military to reconsider using aluminum for some of its particularly weight critical designs.

Many ships are constructed of high-strength steels or use high-strength steels in certain locations. Some of the high-strength steels are twice as strong (yield strength) as ordinary steel, yet they do not save much weight in large ships. The reason is that structural details composed of high-strength steels have almost the same fatigue allowable stresses as ordinary steel, and, hence,

these ships require just as much material to resist hull girder bending. The extra strength can only be used to resist secondary loads. High strength steels can be effectively utilized for the primary structure in small crafts and in secondary structures governed by secondary loads to reduce structural weight compared to ordinary steel construction. Improvement in the fatigue characteristics of high-strength steels is necessary to significantly improve the structural weight fraction of larger vessels.

4.3.1.1 Composites

Composite structures consist of fiber reinforcements (such as E-glass or carbon) encapsulated in a resin matrix (such as vinyl ester or phenolic). Composite materials can be used to produce single-skin, stiffened, or sandwich structures; see Figure 4.3.1-2. They have been used for primary structures on small craft or vessels for many years. They are also applicable for secondary structures such as decks, foundations, doors, hatch covers, enclosures, deckhouses, stacks, and masts.

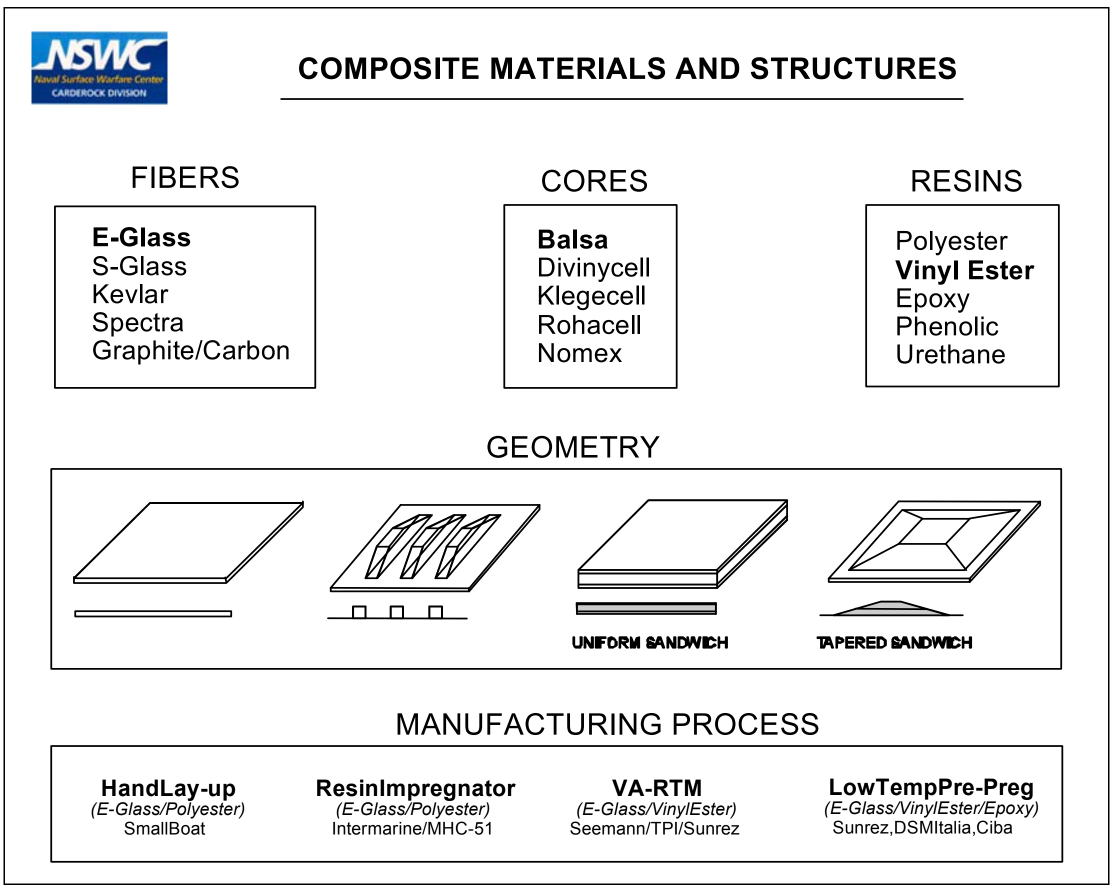

Figure 4.3.1-2: Composite Materials and Processes

4.3.1.2 Titanium

Titanium has a yield strength higher than most high-strength steels, with a density of only 57 percent that of steel. The potential weight savings exceed that of aluminum and it has much better fire resistance. Titanium alloys have been used extensively in the aerospace industry in the United States and have received some attention in the automotive industry. Timetal 10-2-3 (Ti-10V-2Fe-3Al) is used in the main landing gear of the Boeing 777. Timetal 15-3 (Ti-15V-

3Cr-3Sn-3Al) has been used in environmental control system ducting, firefighting bottles, door springs, and small nut clips. This alloy has good formability and was used for more than one hundred formed parts on the B1B bomber. The superplastic alloy SP-700 has been used in place of stainless steel in steam turbine blades, hand tools, and golf club heads.

Titanium has been used for submarine hulls in the former Soviet Union. The biggest issue with titanium is its cost and availability. Titanium is thirty times more expensive than steel and it must be imported from Russia. In addition, it is more difficult to weld.

4.3.2 Technology Goals

In the struggle to develop a low-cost, high-strength/lightweight material, several obstacles remain. Stiffness and fire performance are issues that must be addressed. Effective repair procedures that ensure structural integrity must also be developed. Material development costs can be significant and consideration must be given to production-mode acquisition costs. Many of the materials require strict environmental control during fabrication, requiring significant capital investments in infrastructure development.

The increased use of aluminum as the primary structural material for larger and larger ships has improving our understanding of the structural behavior of large aluminum structures. This evolution has already increased our ability to produce a more efficient and robust design. The U.S. military is currently evaluating the suitability of some of these commercial designs for military operations. It appears that significant modifications to local details must be made to existing, restricted operational commercial designs, and a better understanding of the loads and the fatigue behavior of the newer aluminum alloys is needed before they can be considered for unrestricted, military operations.

Conventional composite fabrication processes are critical in quality control. Because of this issue, new fabrication processes (see Figure 4.3.2-1) such as vacuum-assisted resin transfer methods (VARTM) have been developed to provide more consistent quality control. However, variability in material properties continues to be an issue and is highly dependent on the manufacturing process selected. Worker skill also continues to play a significant role in the quality and consistency of the resulting composite material. Further research and development is needed to develop low-temperature, low-cost/high-quality manufacturing processes and fiber/resin combinations that minimize material property variation and maximize strength and stiffness characteristics.

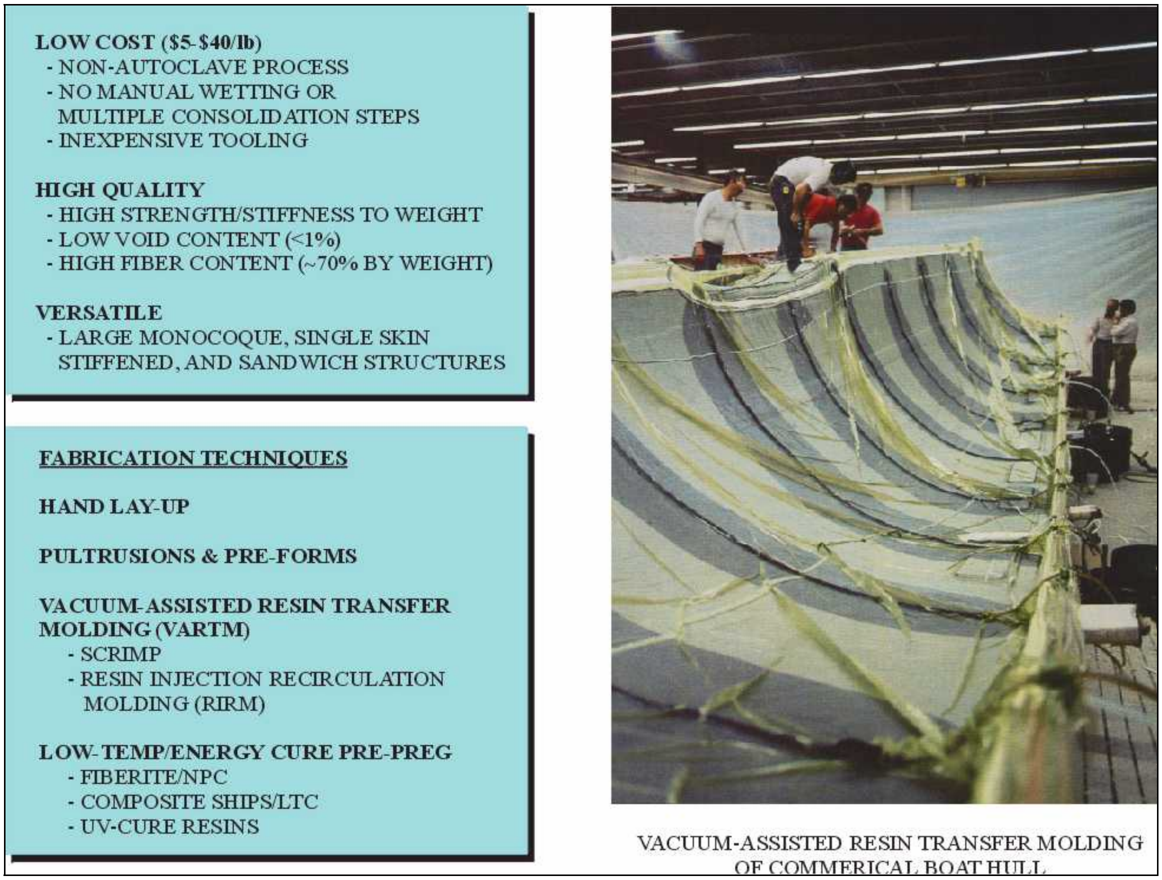

Figure 4.3.2-1: Composite Fabrication Techniques

Stiffness is not as critical for secondary structures or for primary structures when the ship length is less than 130 meters. However, when the hull length starts to exceed 130 meters, stiffness becomes more of a concern for virtually all of the non-steel materials currently under consideration. For the primary hull structure of large ships, the limited stiffness of non-steel materials can yield a large hull deflection, which may be problematic for critical alignments. Maintaining hull girder stiffness may be required to avoid hull resonance issues such as springing and whipping. In the near-term, E-glass and carbon composites are effective in reducing weight in secondary structures and they can also be used for the primary structures of small craft, but they have a low stiffness for the primary hull structure bending in large ships (over 130 meters).

Fatigue characteristics must be improved with many of the material options. The fatigue limitations of aluminum and high-strength steels result from their as-welded properties. Improved welding methods (or eliminating welding by adhesive joining methods) can increase the fatigue allowable stresses for both aluminum and high-strength steels. For example, flush ground welding of aluminum increases the fatigue strength to two-thirds that of ordinary steel, resulting in a fifty-percent structural weight saving. Weight savings for high-strength steels would be proportional to any increases in fatigue allowable stresses from advanced welding/

joining techniques. Such advanced welding and joining techniques need to be investigated and developed, and are certainly possible in the far-term.

The current fatigue database of titanium components is inadequate to ensure a reliable titanium ship design. Fatigue tests of welded titanium components are needed to develop design criteria in order to design a titanium ship in the mid to far-term timeframes. Since titanium is non-magnetic, new non-destructive inspection (NDI) methods must be developed to replace the common magnetics-based inspection methods currently in use for steel.

4.3.2.1 Summary of Material Properties

A summary of the material properties as they currently exist is shown in Table 4.3.2-1. Relative stiffness in Table 4.3.2-1 is represented by Young's Modulus (Modulus of Elasticity).

It is expected that, as composites become more widespread, their unit costs will decrease in the far-term. The following unit costs are raw material costs only; producibility issues and fabrication costs are not included in this study. In general, the structural material costs of a steel ship are very small compared with fabrication, installation, and equipment/machinery costs. The average costs for cutting, welding, rigging, painting, and material is on the order of \$25 to \$30 per pound for a steel combatant and half that for a commercial ship. Thus, in the future, the total fabrication costs would likely be much closer for all of these items than are the material-only costs of Table 4.3.2-1.

4.3.3 Development Plan for Materials

High-speed Naval ships of 500 to 3,000 mt require the further development of low-cost, high-strength/ lightweight materials. Several lightweight/relatively low-stiffness materials are already being used for secondary structures and for the primary structure of ships less than 130 meters in particularly weight-critical applications. Cost considerations often dictate the use of steel construction for components that are not weight-critical.

The material of choice for the primary structure, as ship lengths exceed 130 meters, remains steel. The primary reasons for the selection of steel include cost, stiffness, fatigue performance, fire performance, property variability of certain other material options, and shipyard experience with steel.

Initially, all of the material options being considered for high-speed, small Naval ships must be investigated. Eventually, many of the options will be removed from further consideration because of insurmountable issues that are discovered during the investigations. The "weeding out" process is necessary and unpredictable, and will reduce the number of material options available for certain applications.

Table 4.3.2-1: Material Properties Summary

Material	Density Lb/ft^3	Yield Strength Ksi	Young's Modulus Ksi	Fatigue Stress Ksi	Fire Resistance	1995 Costs $/Lb
ABS Grade A steel	491	34	29,600	20	Good	0.29
ABS Grade AH steel	491	55	29,600	20	Good	0.34
Aluminum (5086-H34)	166	16-22	10,000	10	Poor	1.65
Titanium	280	140	16,500		Fair	10.00
Sandwich Panel-LASCOR (stainless steel)	245-320	55	29,600	20	Good	
Composite Resins - Vinyl Ester - Phenolic - Epoxy	 70 72 75	 11-12 5 7-11	 490 530 530		 Poor Good	 1.74 1.10 3.90
Composite Fibers - E-glass - S-glass - Carbon-PAN - Kevlar 49	 162 155 110 90	 500 665 350-700 525	 10,500 12,600 33-57,000 18,000			 1.14 5.00 12.00 20.00
Composite Cores - Balsa - Honeycomb Nx HRH-78	 7 6	 1.3 N/A	 370 60	 N/A N/A	 Insulator	 3.70 13.25
Composite Laminates - Solid Glass/Polyester - Solid Glass/Vinylester - Solid Carbon/Epoxy	 96 90 97	 20 50 88	 1,400 3,000 8,700	Excellent fatigue life	Poor Fire Resistance – Good Insulator	 2.50 3.50 10.00
Composite Sandwich - Glass/Poly Balsa Sandw. - Glass/VinE PVC Sandw. - Carbon/Epoxy Nomex	 24 18 9			Excellent fatigue life	Poor Fire Resistance – Good Insulator	 4.00 5.00 20.00

Although it is often difficult to separate material developmental issues from structural developmental issues, a summary of the research and development effort was shown earlier in Figure 4.1-3 and includes:

- The development of improved welding and joining technology for improved ultimate strength and fatigue performance (all materials).
- The development of cost-effective, structurally-sound repair procedures (all materials).
- Research to improve material stiffness characteristics (all materials).
- Detailed cost-benefit analysis identifying acquisition and life-cycle trends.
- The development of low-cost methods to meet fire containment and toxicity criteria (all materials).
- The further development of a reliable, low-temperature, curing process (composite materials).
- The development of a manufacturing process and material composition that yields high strength while ensuring consistent material properties (composite materials).
- The development of high-strength/stiffness fibers and resins (composite materials).

4.4 Structural Concepts

4.4.1 State-of-the-Art

Ordinary steel and high-strength steel stiffened panels continue to be the standard for large ship primary and secondary structures. LASCOR and composites are increasingly being used in secondary structures to reduce weight when necessary. These technologies are currently under investigation for use in the primary structures of large vessels.

For ships less than 130 meters in length, aluminum plate-stiffener construction and composite construction are the choices for the primary and secondary structures when minimum weight must be achieved. Fatigue strength of the aluminum vessels has been an issue, and careful monitoring is of importance.

4.4.1.1 Sandwich Metals (LASCOR – laser-welded corrugated core)

Sandwich metal structures consist of two thin face sheets of metal joined together by a corrugated core; see Figure 4.4.1-1. The separation of the face sheets provides high bending stiffness at a low weight. Stainless steel LASCOR panels have been used on Navy ships for over a decade to save weight for platforms, hanger doors, and deckhouse enclosures; see Figure 4.4.1-2 for past and proposed LASCOR applications.

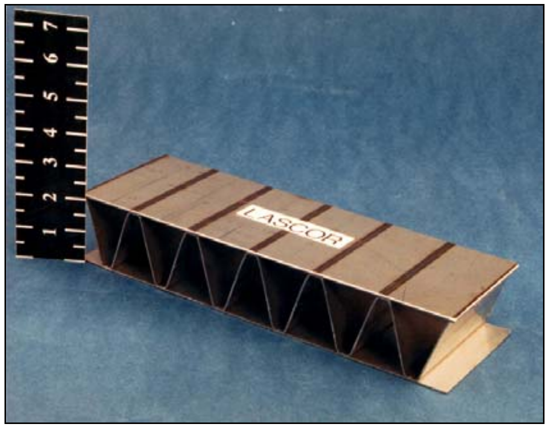

Figure 4.4.1-1: LASCOR Technology

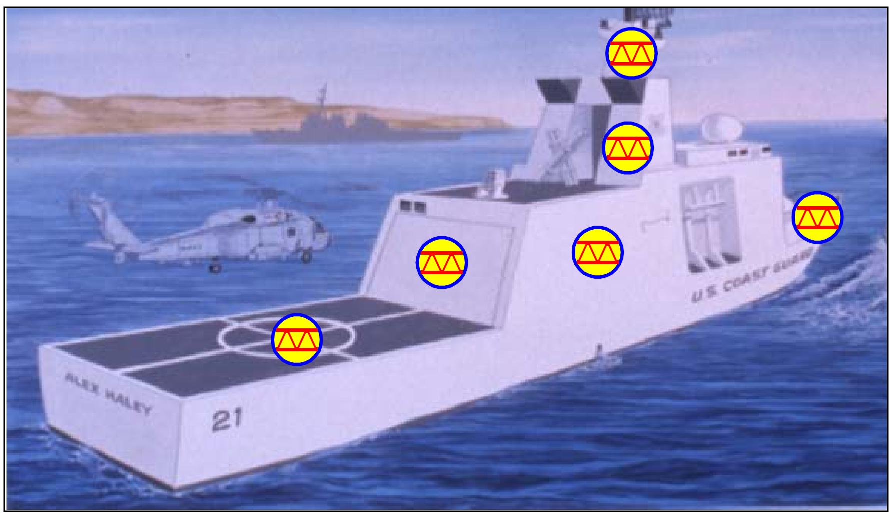

Figure 4.4.1-2: Proposed LASCOR Applications

Sandwich metal structures have a number of advantages over conventional steel construction:

1. Compared to conventional steel structures, metallic sandwich structures have reduced weight and increased stiffness. They are ideal for secondary structures such as internal decks, ramps, hatch covers, bulkheads, and deckhouses, with weight savings of 20 to 50 percent over conventional steel construction.
2. They result in reduced fabrication and outfitting costs. LASCOR panels are 20 percent cheaper to build and install than steel grillages. They have a high dimensional stability that helps reduce assembly and fit-up costs in the shipyard. Outfitting of distributive systems and installation of insulation costs are also reduced because of the smooth surfaces resulting from the elimination of most of the stiffeners.
3. The elimination of stiffeners on decks and bulkheads increases the usable volume within the total ship.

4. Metallic sandwich panels have improved thermal and fire performance. The space within the core offers inherent thermal insulation and protection against the spread of fires.
5. The high stiffness of sandwich panels reduces vibrations. Panels can also be sound-isolated from surrounding structures.
6. Laser-welded sandwich panels are ideal applications of automated fabrication techniques. They can be pre-fabricated as panels at high-efficiency factories before shipboard installation in the shipyard.

One of the issues with sandwich structures is corrosion protection of the voids within the core. Stainless steelstructures, currently used in the fleet, are one solution. They have suffered no corrosion or fatigue damage after a decade of service. Another solution, which has been successfully tested in the field with ordinary steels (carbon steels), is to fill the void spaces with foam. Although our experience is limited to steels, sandwich panels can also be made from corrosion-tolerant metals such as aluminum or titanium. Such lightweight materials can further reduce structural weight and would be available for the mid to far-term applications.

4.4.1.2 Composite Structures

Composites have been used as the primary structure for small vessels for many years. They have also been used for secondary components. Below, in Figure 4.4.1-3, is a summary of the composite applications available.

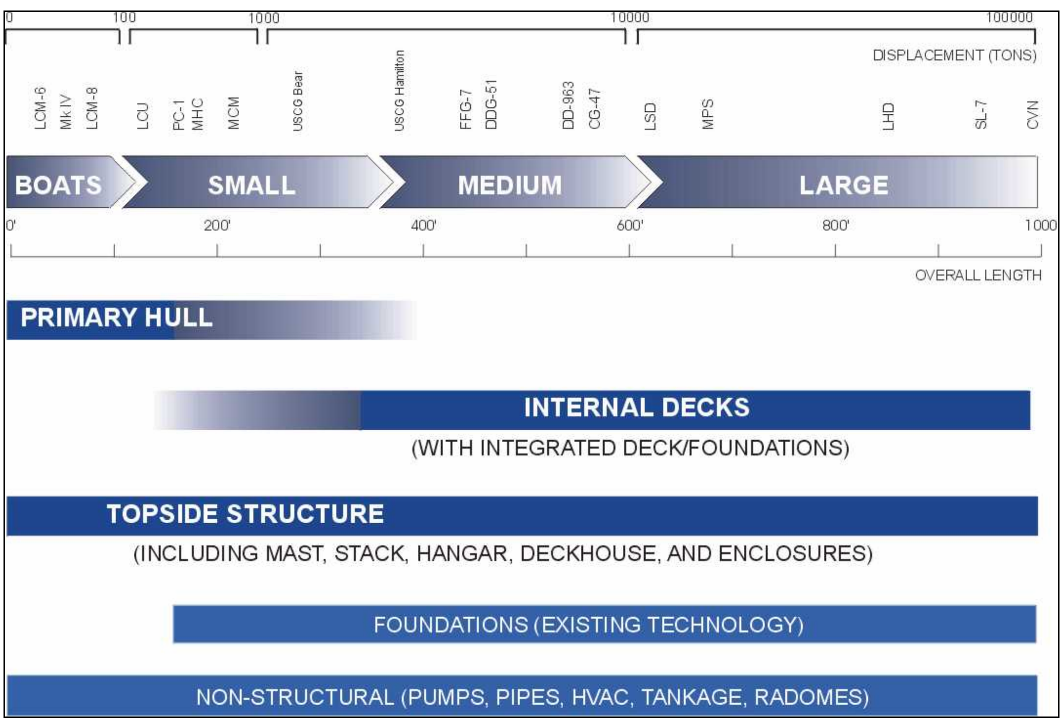

Figure 4.4.1-3: Composite Applications

Composites offer many advantages compared to standard metallic structures:

1. They are lightweight. Weight reductions of 35 to 50 percent, compared to steel, can currently be realized for secondary structures made of E-glass composite laminates. Since secondary structures comprise a significant fraction of the total structural weight, this translates into a worthwhile savings in total ship weight.
2. Composite structural elements have better dimensional stability than steel elements. This is an aid to the fit-up and assembly in the shipyard, and results in lower fabrication costs and better overall dimensional tolerances.
3. They have reduced noise and vibration properties. Composites have inherently better damping and compliance than metallic structures. They also have the potential to be adapted into smart structures, i.e., structures that can monitor and/or alter their properties in service
4. Fires are more easily contained in composite structures because of their low thermal. conductivity. The cores in composite sandwich panels are good thermal insulators
5. The designer has increased flexibility to tailor the composite structure to the particular need. Complex geometries can be designed to optimize the strength and stiffness, or to enhance producibility by minimizing the number or location of joints.
6. Composites have lower life-cycle maintenance costs than steel structures. Fewer inspections, less painting, and fewer repairs are needed over the life of the ship because of the non-corrosion and reduced fatigue damage of composites over metallic structures.

Tables 4.4.1-1 through 4.4.1-4 (Reichard, 1988) present the relative weights of panels having equal stiffness and equal strength under both in-plane (axial) and bending loads. Composites are more advantageous than steel or aluminum when compared on an equivalent strength basis rather than on stiffness basis.

Table 4.4.1-1: Panels of Equal In-Plane Stiffness*

Material	Skin Thick. (inch)	Core Thick. (inch)	Elastic Modulus (ksi)	Weight (lb/sqft)
Steel	0.08	0	30,000	3.36
Aluminum	0.25	0	10,000	3.62
E-Glass (0,90)	1.14	0	2,200	9.99
Kevlar (0,90)	0.60	0	4,200	4.49
Carbon (0,90)	0.35	0	7,200	2.87
Uni-E-Glass	0.57	0	4,400	4.99
Uni-Kevlar	0.30	0	8,400	2.24
Uni-Carbon	0.17	0	14,400	1.43
E-Glass/Core (0,90)	0.57	5	2,200	15.15
Kevlar/Core (0,90)	0.30	3	4,200	7.59
Carbon/Core (0,90)	0.17	1.75	7,200	4.68
Uni-E-Glass/Core	0.28	3	4,400	8.09
Uni-Kevlar/Core	0.15	1.5	8,400	3.79
Uni-Carbon/Core	0.09	1	14,400	2.47

* All panels have a stiffness of 2.5×10^6 pounds/inch

Table 4.4.1-2: Panels of Equal In-Plane Strength*

Material	Skin Thick. (inch)	Core Thick. (inch)	Yield Strength (ksi)	Weight (lb/sqft)
Steel	0.19	0	80	7.56
Aluminum	0.26	0	58	3.74
E-Glass (0,90)	0.34	0	44	3.00
Kevlar (0,90)	0.25	0	60	1.89
Carbon (0,90)	0.14	0	105	1.18
Uni-E-Glass	0.17	0	88	1.50
Uni-Kevlar	0.13	0	120	0.94
Uni-Carbon	0.07	0	210	0.59
E-Glass/Core (0,90)	0.17	1.75	44	4.80
Kevlar/Core (0,90)	0.13	1.25	60	3.18
Carbon/Core (0,90)	0.07	0.75	105	1.96
Uni-E-Glass/Core	0.09	1	88	2.53
Uni-Kevlar/Core	0.06	0.5	120	1.46
Uni-Carbon/Core	0.04	0.5	210	1.11

* All panels have a maximum strength of 15.0 x 10^3 lbs/inch width

Table 4.4.1-3: Panels of Equal Flexural Stiffness*

Material	Skin Thick. (inch)	Core Thick. (inch)	Mom. of Inertia (inch^4)	Weight (lbs/sqft)
Steel	0.74	0	0.0335	29.74
Aluminum	1.06	0	0.1004	15.39
E-Glass (0,90)	1.76	0	0.4543	15.46
Kevlar (0,90)	1.42	0	0.2386	10.71
Carbon (0,90)	1.19	0	0.1387	9.80
Uni-E-Glass	1.40	0	0.2262	12.25
Uni-Kevlar	1.13	0	0.1196	8.51
Uni-Carbon	0.94	0	0.0697	7.79
E-Glass/Core (0,90)	0.23	2	0.4539	6.04
Kevlar/Core (0,90)	0.16	1.75	0.2380	4.15
Carbon/Core (0,90)	0.15	1.375	0.1385	3.83
Uni-E-Glass/Core	0.15	1.75	0.2272	4.41
Uni-Kevlar/Core	0.13	1.375	0.1185	3.31
Uni-Carbon/Core	0.11	1.125	0.0692	2.96

* All panels have a stiffness (EI) of 1.0 x 10^6 pound-inch^2

Table 4.4.1-4: Panels of Equal Flexural Strength*

Material	Skin Thick. (inch)	Core Thick. (inch)	Yield Strength (ksi)	Weight (lbs/sqft)
Steel	0.19	0	80	7.56
Aluminum	0.26	0	58	3.74
E-Glass (0,90)	0.34	0	44	3.00
Kevlar (0,90)	0.88	0	17	6.65
Carbon (0,90)	0.14	0	105	1.18
Uni-E-Glass	0.17	0	88	1.50
Uni-Kevlar	0.44	0	34	3.33
Uni-Carbon	0.07	0	210	0.59
E-Glass/Core (0,90)	0.12	1.25	44	3.47
Kevlar/Core (0,90)	0.20	2	17	5.09
Carbon/Core (0,90)	0.09	0.75	105	2.19
Uni-E-Glass/Core	0.09	0.875	88	2.46
Uni-Kevlar/Core	0.15	1.375	34	3.61
Uni-Carbon/Core	0.05	0.625	210	1.52

* All panels have a maximum moment capacity of 7.5x10^2 foot*pounds

There are a number of issues associated with composites:

1. Flammability, smoke, and toxicity dangers are the main concerns associated with composites. They are handled in several ways. For unmanned spaces in secondary structures not subject to severe fire threat, a thin thermal barrier coating, no coating, or passive fire protection may be used. For manned spaces in secondary or primary structure, thermal protective insulation is used (see Figure 4.4.1-4).

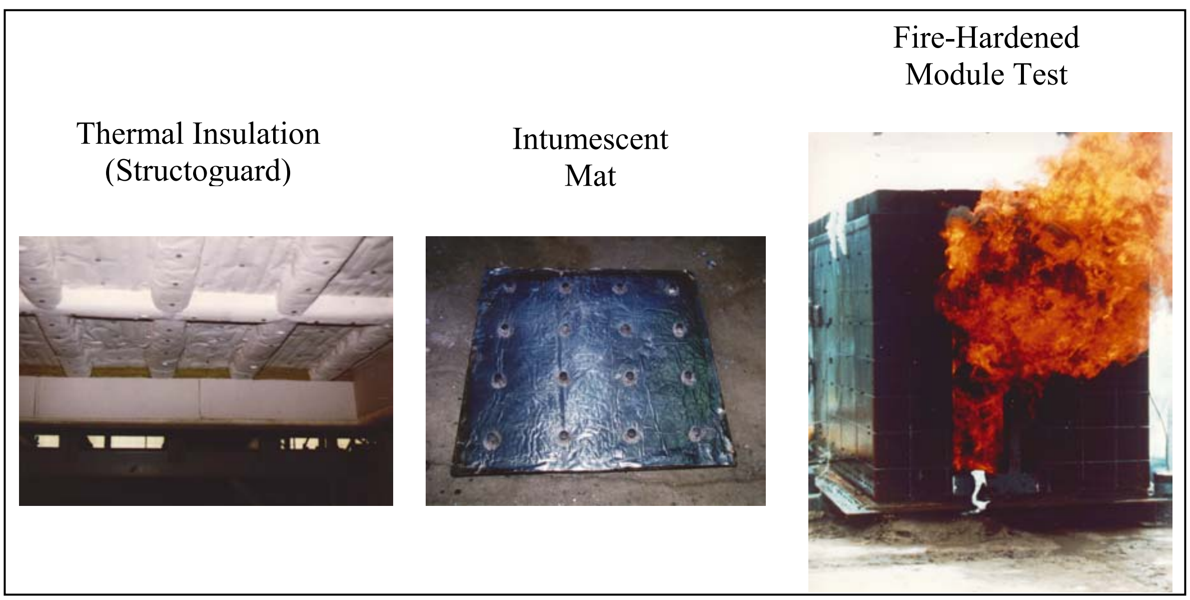

Figure 4.4.1-4: Composite Fire Protection

2. Composite designs are normally limited by stiffness, not strength. For hybrid structures, such as a composite deckhouse on a steel hull, the lower stiffness results in lower stresses and better fatigue performance. However, for the primary hull structure of large ships, the low hull overall stiffness may be a problem for deflections limits of conventional propeller shafts. Other propulsors (such as waterjets, electric drive, and podded propulsion) may render this issue moot.

3. There are limited design data and analytical tools. Design data and tools are becoming increasingly available for more common materials (e.g., glass polyester or vinylester) and for structural configurations, joints, and fabrication processes. However, in most cases, experimental validations are still needed.

4. There is minimal shipyard experience for constructing large composite ships. The largest composite ship hulls are those of naval minehunters and minesweepers, with lengths of 50 to 60 meters.

4.4.2 Technology Goals

There are a number of issues that must be explored before LASCOR and composite structural technologies are considered for primary and secondary structural applications for small high-speed Naval ships. Obviously, because LASCOR and composite structural concepts have been demonstrated for secondary applications, the research necessary to implement them for high-speed combatant secondary applications is not as extensive as for primary structure applications, and could be accomplished in the near-term. There are several issues to be resolved before either LASCOR or composites can be considered for primary structural applications, hence, the earliest they could be available for primary structural consideration would be in the far-term applications.

4.4.2.1 LASCOR

Although LASCOR has been used in commercial and military secondary structural applications, several issues need to be addressed before it can be reliably used for secondary and primary structures of HS combatant ships.

As mentioned earlier, one of the issues with metallic sandwich structures is corrosion protection of the voids within the core. As discussed, stainless steel has already been used in the fleet to eliminate this problem in several applications and has suffered no corrosion or fatigue damage after a decade of service. Another solution, which has been successfully tested in the field with ordinary steels (carbon steels), is to fill the void spaces with foam. These results have been promising. Our mid to far-term goal is to develop sandwich panels from lightweight, corrosion-tolerant metals such as aluminum or titanium and optimize the sandwich panel characteristics such as structural weight, acquisition costs and life-cycle costs.

Other issues associated with manufacturing will be resolved as the shipyard experience increases with metallic sandwich panels. Efficient repair procedures need to be further developed and optimized. Draft design guides and standards exist, but must be formally documented and approved by the Navy and regulators for commercial applications. The fatigue performance of metallic sandwich panels must be further defined and validated for both primary and secondary loads along with full-scale structural static tests. This will allow a reduction in the factors of safety now assumed, resulting in lighter and more reliable structures.

In the far-term, techniques to form complex shapes, not just flat or singly-curved panels, must be developed and optimized. The ability to form hybrid metallic sandwich structures also has far-term potential for weight and cost reductions.

4.4.2.2 Composite Structures

Similar to LASCOR, composites have been used in commercial and military secondary structural applications, but several issues need to be addressed before they can be reliably used for primary structures of high-speed Naval ship.

The Norwegian patrol boat *KNM Skjold,* and the Swedish combatants *Smyge* and *Visby* have demonstrated composite technological advancements for small high-speed combatant applications. Advanced composite materials and integrated structural systems were used for these hullforms, in part, for their ability to improve the signature characteristics and shock resistance of the structure. In addition, the reduced weight and simplified construction and outfitting demonstrated during the construction of these small vessels are important attributes in a small, high-speed naval ship.

The recently-commissioned fast patrol craft *KNM Skjold*[1] (see Figure 4.4.2-1) is an SES hullform with an overall length of 47 meters and displacement of 260 mt. Fiber-Reinforced Plastics (FRP) sandwich construction is used throughout the vessel, with vinyl ester or polyester resins and either a PVC or PMI core material. In locations requiring high stiffness, carbon fibers were used; otherwise, E-glass was chosen for the laminates.

The Swedish corvette *Visby*[2] (see Figure 4.4.2-1) is also one of the more advanced combatants using composite technology. With an overall length of 73 meters, the *Visby* is the largest commercial or combatant ship hull entirely constructed of Carbon Fiber-Reinforced Plastics (CFRP). The *Visby* was built using a vacuum-infused process consisting of sandwich construction with a PVC foam core and vinyl ester resin.

These small warships demonstrate the flexibility and capability of composite technologies for small surface combatants. Further research to improve material properties, joints and connections, and shipyard producibility is necessary before these technologies are matured for primary hull structures of larger ships.

[1] http://www.knmskjold.org/english, 2000.
[2] http://www.naval-technology.com/projects/visby, 2001.

Figure 4.4.2-1: Composite Small Combatants

Research has shown the potential of multi-layered, balsa-cored sandwich structures for containing fires and preventing structural collapse or excessive deflections. The French surface combatant *La Fayette* uses balsa-cored sandwich construction on the deckhouse for this purpose. Also, the potential use of phenolics and other fire-retardant resins has been demonstrated for small fires burning for 20 to 30 minutes. These potential fire performance improvements must be further developed.

Although joining technology is very critical for all composite structural applications, it is particularly important when considering composite materials for primary structure. There is a significant reduction in in-plane strength characteristics at the joints of composite structures. This problem can be eliminated or reduced if a monolithic rather than modular construction process is adopted. However, for very large ship lengths, it would seem that modular construction would be required, and a significant research and development effort would be needed to improve composite joining technology.

There are limited design data and analytical tools available. Design data and tools are becoming increasingly available for more common materials (e.g., glass polyester or vinylester) and for structural configurations, joints, and fabrication processes. However, in most cases, further development and experimental validations are still needed.

Inspection and repair methods are generally available for most composite structures. However, inspection can become difficult for thick, sandwich structures. Therefore, some of the current non-destructive evaluation (NDE) methods must be further developed.

4.4.3 Projected Weight Savings

Table 4.4.3-1 is a summary of the projected weight savings for the various materials in the near, mid, and far-terms. All of the weight savings are relative to ordinary steel (ABS grade A) of conventional stiffened plate construction. The percentage reductions are applied to the entire structural weight (SWBS 100) of the ship.

Table 4.4.3-1: Summary of Weight Savings (Percent)

Material	Near-Term	Mid-Term
Aluminum	30	30 - 40 with new alloys
Titanium & Advanced Metals	High Risk	40 - 55 secondary structure, 15 overall
Metal Sandwich (LASCOR)	35 - 50 secondary structure, 10 overall (steel)	40 - 55 secondary structure, 15 overall (steel)
Composites (300' ship length)	20 - 40 with Glass or Carbon fibers	30 - 45 with Glass or Carbon fibers

4.4.4 Development Plan for Structures

Small high-speed Naval ships require not only the further development of low-cost, high-strength/lightweight materials, but also the most cost and weight efficient structural concepts that can implement these material enhancements. For secondary structures and for the primary structure of ships less than 130 meters, several lightweight/relatively low-stiffness materials are already being used in particularly weight-critical applications. Cost considerations often dictate the use of steel construction for components that are not weight-critical.

Sandwich metal structures such as LASCOR have been used for non-primary load carrying structure and have been effective in reducing structural weight. To optimize LASCOR secondary structures and strength decks for high-speed, small naval ships, additional research and development would be needed. In the near-term, these efforts would include:

- The development of ways to reduce or eliminate corrosion within the void spaces such as using alternate lightweight metals as the primary material or filling the void spaces with foam.
- The further development of efficient repair procedures and inspection techniques.
- The documentation and approval of design guidelines.
- Testing and evaluating primary load carrying capacities and fatigue performance.
- The optimization of cargo decks.

To optimize composite secondary structures and strength decks for high-speed, small Naval ships, additional research and development would also be needed in addition to the material developmental effort described earlier. These efforts include:

- Significant testing, analysis, documentation, development of design tools, and approval of design guidelines.
- Joint detail development.
- The further development of efficient repair procedures and inspection techniques.
- Testing and evaluating primary load carrying capacities.
- The development of cores and materials for fire containment.
- The optimization of cargo decks.

Although composites have been used for primary structural applications in vessels below 130 meters, there is no experience with its use for large vessels, and a significant development effort would be required. In addition to the model tests discussed earlier to determine loads, full-scale demonstrations verifying large-scale joining technologies, manufacturing processes, and at-sea performance would be needed.

4.4.5 Summary of Required Technology Development

Table 4.4.5-1 is a summary of the technologies to be investigated and developed for use in the near, and mid-term structures of small high-speed Naval ships.

Table 4.4.5-1: Technology Development Needs

Material	Near-Term	Mid-Term
Aluminum	State-of-the-Art for ship lengths < 300', untried >300'	new alloys with better fatigue properties
Titanium & Advanced Metals	no time for multi-year R&D effort	define fatigue & strength properties
Metal Sandwich (LASCOR)	approved design standards & rules (steel)	validate fatigue properties & improve corrosion resistance
Composites (300' ship length)	State-of-the-Art for ship lengths < 200', better fire resistance	improved & validated design tools

4.5 Preparation of ABS Guide for High Speed Naval Craft

Classification is the process of verifying that the hull, machinery, and electrical systems and related components meet technical requirements for fitness, safety, and environmental soundness. These technical requirements are contained in Rules that are developed by the

classification society. The vessels are verified to comply with rule requirements in their original design plans, as constructed, and throughout their operational life. The set of Rules to which a vessel is designed varies depending on its type of classification and service, as well as any special notations; for example, many high-speed ferries are Classed under High-Speed Craft Rules. ABS is currently in the process of developing *The ABS Guide for Building and Classing High Speed Naval Craft* and *Rules for Building and Classing Naval Vessels* for warships and vessels engaged in military missions. For a Naval ship design to fall under the ABS High Speed Naval craft rules, the maximum ship speed must correspond to a Froude Number of 0.39, or greater. In addition, the ship length must be less than 130 m for a monohull, less than 100 m for a multi-hull, and less than 90 m for an SES.

A process was developed to ensure that lessons from experience would not be lost in the development of Naval Vessel Rules. This process included a comparison of naval and commercial standards that led to an initial draft set of standards. These draft standards were then reviewed and modified by technical committees and industry, but have not been reviewed or approved by the Navy. Finally, provisions were made for annual updates of the standards.

At the current time, the industry has little experience designing the hullforms being considered for the high-speed Naval missions. Therefore, to prepare for an initial draft set of standards, significant research as described previously in loads, materials and high-strength/lightweight structures development is necessary. Because a set of standards ensuring the fitness, safety, and environmental soundness for high-speed Naval vessels is the ultimate goal, ABS involvement in the research and developmental efforts will ensure that they will be provided with all the necessary information to class the vessels.

4.6 Loads, Materials and Lightweight Structure References

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5.0 MACHINERY SYSTEMS

5.1 Introduction

Significant extension of machinery technology is required for 50 to 60 knot, 3,000- mt naval ships. Propulsion machinery must be compact, lightweight, and fuel-efficient, yet produce and transmit very high levels of power.

5.2 Prime Movers

Prime movers for small, high-speed naval ship designs range in power from current technology turbines producing about 10 MW, to large near-term turbines producing up to 43 MW. Existing gas turbines with ratings of up to 32 MW are adequate for most high-speed, small combatant missions. Only the largest and fastest designs show a requirement for a near-term nominal 43 MW turbine. Navy certification testing and, possibly, some technology development will be required to make either a 32 MW or 43 MW gas turbine a near-term option.

5.2.1 State-of-the-Art

The General Electric LM2500 gas turbine, at a Navy-certified maximum continuous power rating of 19.6 MW (26,250 HP) at 100^{o}F, is widely used in a large number of U.S. Navy ships. The LM2500 is an aero-derivative gas turbine that was directly derived from GE's CF6 family of commercial aircraft engines and GE's TF39 military engine. There are over 800 LM2500 gas turbines in service in more than 24 international navies.

GE LM2500+ turbines are currently in service powering waterjets in the 42-knot ferry Corsaire 13000 Liamone with the turbine rated at 25 MW. The turbine is also in service in the cruise liner Millennium in an integrated electric plant. The first military application of the LM2500+ will be in the LHD Wasp-class large-deck multi-purpose amphibious assault ship. The LM2500+ has been certified to have a U.S. Navy rating of 26.1 MW (35,000 shp) for the LHD application.

In late 2003 Rolls-Royce will begin delivering Marine Trent 30 gas turbines, which are expected to have a power rating of about 32 MW (41,500 bhp) at 100 deg. F. The MT30 shares 80 percent commonality with the Trent 800 aero engine of which more than 500 have been sold or ordered.

Gas turbine R&D advances have resulted in some simple-cycle plants such as the MT30 operating with efficiencies of more than 40 percent at full power, but with significant reductions in fuel efficiency at part power. Turbines using more complex cycles exploiting intercooling and recuperation (ICR) technologies reportedly achieve specific fuel consumption rates closely approaching the very flat curve characteristic of larger diesel engines. Warships may be the first to benefit from ICR technology in the form of the ICR-based Northrop Grumman/Rolls Royce 25 MW WR21 marine gas turbine, an engine derived from the aero engines (RB211 and Trent) combined with intercooler and recuperator systems. This engine has successfully completed a 500-hour land-based endurance test in a joint US/UK/France program. The WR21 is more fuel efficient than simple-cycle gas turbines across its entire power range. A maximum reduction of about 30 percent is achieved at the bottom of the power range, with savings of about a quarter

that at full power. The WR21 turbine has been ordered for use in the UK Type 45 destroyer in 2007. The size and complexity of the WR21 ICR gas turbine and the LM2500 turbine are illustrated in Figure 5.2.1-1.

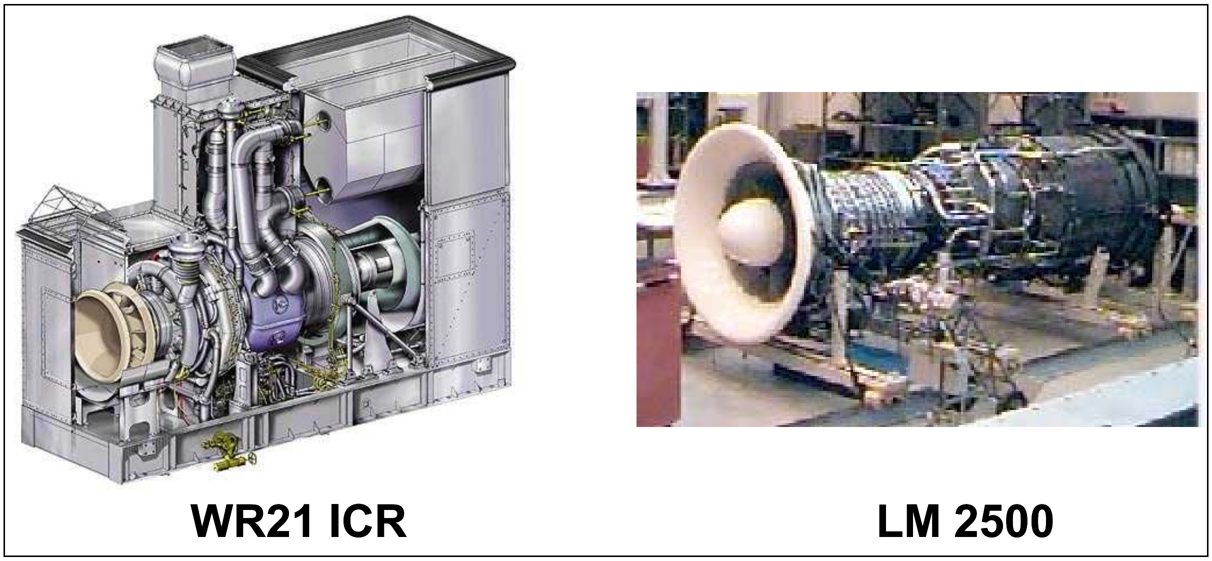

Figure 5.2.1-1: WR21 ICR and LM2500 Gas Turbines

GE Marine Engines' uprated LM6000 aero-derivative industrial gas turbine is another developmental choice for marine propulsion application. Although the LM6000 is designed for offshore use, environmental requirements are similar to those of marinized turbines. The major development issue would be modifications to the control system to be compatible with propulsor loading requirements. The LM6000 uprated models offer over 40 MW (50,000 shp) with 42 percent ISO thermal efficiency. Currently, there are 115 LM6000s in operation.

The development of the FastShip Atlantic project resulted in a design for a 40-knot, 1420 TEU commercial vessel powered by five 50 MW (68,000 bhp) gas turbines. Machinery proposals were submitted by GE and Rolls Royce. GE Marine offered a marine version of its industrial 42 MW (57,100 bhp) LM6000, while Rolls Royce offered the 47.5 MW Marine Trent 50 derived from its aero 800 Trent turbine. While both are untried in the marine environment, they were found to be technically acceptable for the project. The Trent 50 was selected to power the FastShip design, although construction has not begun. Full-power simple-cycle efficiency is 42 percent for the turbine.

The specific fuel consumption rate and power of existing and developmental gas turbines is presented in Figure 5.2.1-2. The figure shows that near-term goals can be satisfied by marinized LM6000/Trent 50 engines.

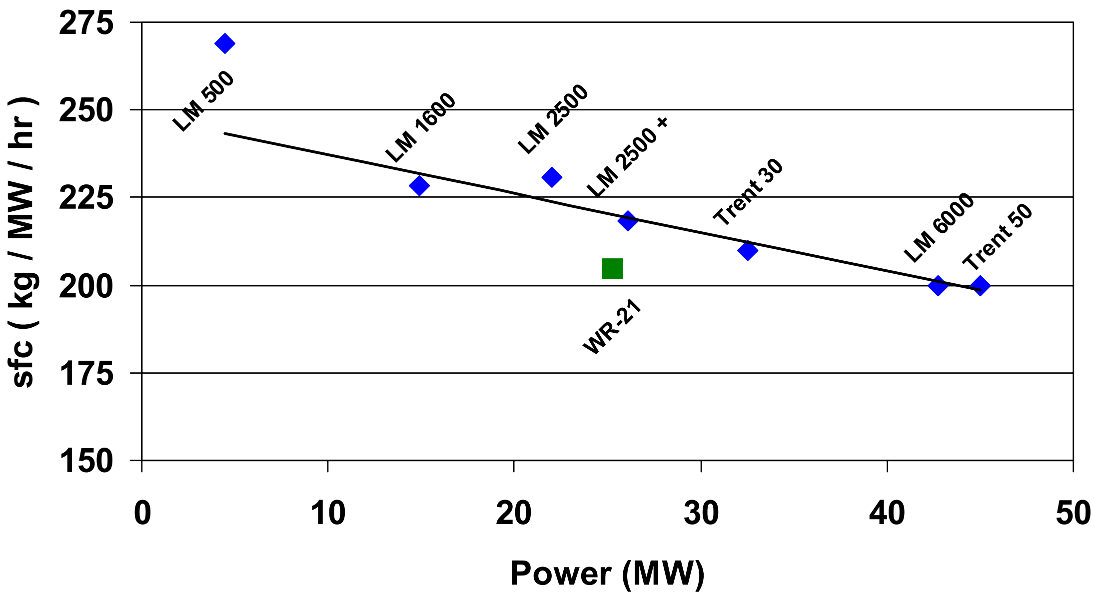

Figure 5.2.1-2: Marine Gas Turbine Technology

5.2.2 Technology Goals

The near-term gas turbine technology goal is development of a marinized 43 MW gas turbine with SFC of 200 g/kWh, very similar to the turbines required for the FastShip Atlantic project. Such a gas turbine has essentially the same turbine performance requirements as for the commercial FastShip Atlantic project. The FastShip Atlantic project experience indicates that the commercial approach to achieve this objective is to develop marinized versions of the LM6000 or Rolls Royce Trent industrial turbines. Development of a near-term turbine requires 3-4 years.

5.2.3 Overview of Development Plan

Development of marine gas turbines with the power required for small HS naval ships is simpler due to the existence of previously-developed industrial, offshore, and aero engines. Existing engines such as the LM6000, and Rolls Royce Trent produce adequate power for near-term 43 MW turbine requirements, but may require marinization. Market forces such as the FastShip Atlantic project may lead to commercial development of these marine turbines independent of military investment. Full-scale fabrication and testing is not required since most of the units have been installed on land or offshore bases and have many hours of operation. Only the qualification of these units for Navy use is required.

The tasks, time to complete each task, and costs associated with developing the needed gas turbine technology are shown in Figure 5.2.3-1. Costs shown are engineering estimates, based on the expected scope of testing and facilities required.

Years	1	2	3	4	5	Funding ($K)
Identify Power Concepts						50
Qualify Trent 30 (32.5-34 MW)						
- Contract						
- Setup Test facility						4,000
- Certification Testing						6,000
Qualify LM6000/Trent 50 (43-50 Mw)						
- Engine Marinization						
- Contract						
- Setup Test facility						4,000
- Certification Testing						6,000
Funding ($K)	50	10,000	10,000	0	0	20,050

Figure 5.2.3-1: Marine Gas Turbine Technology Development Plan

5.3 Waterjets

5.3.1 Introduction

Currently, waterjet propulsion is the preferred propulsion system for high-speed ships above 500 mt. Today's most powerful waterjets and near-term planned upgrades for commercial interests are adequate for small high-speed Naval vessels, but the large transom dimensions required to accommodate the most common type of waterjet may degrade speed-power performance and seakeeping. Nonetheless, because of current availability and depth of design experience, only waterjet propulsion is considered for the present technology development plan.

Two other categories of high-speed propulsors that could be considered as viable candidates for propulsion of 500 to 1,000 mt ships at speeds up to 50 or 60 knots. The two categories are supercavitating (fully cavitating) propellers and partially submerged (surface-piercing) propellers.

Supercavitating propellers employ blunt base foil section shapes that develop lift on the blades' pressure-sides only. These propellers operate best when the blades' suction-sides are completely covered by long, stable supercavities that extend from the sharp nose to beyond the trailing edge, a condition that is achievable only at relatively high speeds. Kruppa (1992) has pointed out that given the intrinsic characteristics of these propellers, the regime of application is for ship speeds of 40 knots and above. At low and medium speeds, the blades of this type of propeller will operate only partially cavitating, with wetted blunt-based blade trailing edges and fluctuating cavity lengths, and thus with relatively poor propulsion performance. The actual high-speed performance of fully cavitating propellers can be impressive. Propulsion of the 320-mt Navy

hydrofoil craft AGEH-1 at a maximum speed of 50 knots was achieved with two 1.58-meter-diameter supercavitating propellers, each capable of absorbing about 13,000 kW (17,400 hp). However, the supercavitating propeller is not a good choice for propulsion of Navy ships that need to operate at a variety of speeds, including cruise and loiter.

Partially submerged propulsors are typically supercavitating propellers that are run at the free surface so that the blades are continuously entering and exiting through the water surface. The suction-side blade cavities are ventilated by the flow of air from above the water. Because of the asymmetric unsteady loading and unloading of the blades of a surface-piercing propeller, there are potential problems with blade-rate vibratory forces, thrust line eccentricity, and large time-average lateral forces. Kruppa (1992) has asserted that if there are a large number of blades, the blade-rate vibration will not be a problem. An important full-scale application of partially submerged propellers was designed for the SES 100B propulsion system. Two controllable pitch, six bladed, 1.07-meter-diameter partially submerged propellers were mounted in tunnels in the vehicle's sidehulls, each capable of absorbing about 5,000 kW at around 1900 RPM. This appears to be the largest application of partially submerged propellers ever built. The SES 100B was able to achieve a maximum speed of 92 knots.

Various configurations of smaller surface-piercing propellers are standard for propulsion of unlimited hydroplanes, offshore racers, and other types of high-speed recreational boats.

Based on the features of attractive propulsion efficiency demonstrated by the SES 100B and the possibility of eliminating the shafts and V-strut appendages of normal propeller installations, a system of partially submerged propellers with subcavitating blade sections was actually investigated as a candidate propulsion scheme for a large monohull combatant ship, and is described in a paper by Rains (1981). A study of propulsion issues using surface-piercing propellers compared with waterjets for one SES ship application was presented by Bjorklund and Allenstrom (1989).

Despite some appealing characteristics, and selected full-scale success, the partially submerged propeller has not yet been developed at a size scale or maturity level needed for direct application in high speed Naval ships.

5.3.2 State-of-the-Art

Large waterjets have basically followed two design approaches; one with mixed-flow and the other with an axial-inducer type blading design. The two primary makers of large waterjet units are KaMeWa and John Crane-Lips. Both of these makers have a long background in the waterjet field and have designs based on non-inducer, mixed-flow type blading.

KaMeWa's largest operational unit is the size 200, with an impeller inlet diameter of 200 centimeters (6.6 feet). The size 180 unit is shown in Figure 5.3.2-1. The 200 size waterjets can be powered by an LM2500^{+} size gas turbine, putting them in the 26 MW power range. Two steerable units of this size are in service on a Corsaire 14000 monohull ferry built by Alstom Leroux. The KaMeWa impeller is a mixed-flow design. Mixed-flow waterjets have radial growth of the blade tip through the impeller and produce a significant radial exit component of

velocity. The maximum impeller diameter can be as much as 40 percent larger than the impeller inlet diameter. It is important for sizing the transom width that the exit housing has a bowl shape beyond this exit blading to accommodate the stator blading and results in installation diameters that are 65-85 percent larger than the inlet diameter.

Figure 5.3.2-1: KAMEWA 180 Waterjet

KaMeWa has performed design work on a size 325 unit (inlet diameter of 325 centimeters, or 10.66 feet) under contract to meet FastShip Atlantic operating requirements of about 40 knots. The size 325 unit would absorb power in the range of 49 MW using a Rolls Royce Marine Trent, but no units have been built to date.

John Crane-Lips has built mixed-flow waterjets that cover up to the 20 MW range. At least four of the largest size, 6-bladed units are in service. Each MEKO-200 class corvette incorporates one LJ 210E reversible Lips booster waterjet, which had to meet shock and noise requirements. Four of these corvettes were built by Blohm + Voss/HDW for the South African Navy.

Rolls Royce/Bird-Johnson is developing the AWJ21 waterjet unit concept. This unit has an advanced mixed-flow impeller mounted in a nacelle arrangement that would be faired with the bottom of the ship hull and incorporates an underwater discharge. The waterjet steering/ reversing equipment would be housed within the nacelle for minimum drag impact. This unit is only in the model development phase, but is intended for up to LM2500 size power applications. The drag of the nacelle for the AWJ21 waterjet concept would probably limit its application to speeds of 40 knots and below.

Axial-inducer waterjets have seen fewer large-scale applications compared to mixed-flow designs. This is due mainly to the significant contraction in the large waterjet market that followed cancellation of the 3KSES program. Proponents of axial-inducer waterjet technology, such as Rocketdyne and Aerojet, discontinued their waterjet businesses about this time. In 1987 Rocketdyne sold their axial waterjet technology and manufacturing business to Kawasaki Heavy Industries. Mixed-flow waterjet manufacturers remained in business by producing smaller units during this waterjet recession and subsequently were able to develop progressively larger mixed-flow waterjets as market demand evolved over the following decades. The belief that inducers

were only applicable to low flow coefficients, and historical data that indicates that the mixed-flow pumps have a hydraulic efficiency advantage over axial designs, also led to the emphasis on mixed-flow designs by other waterjet manufacturers. However, the ability of axial-inducer type pumps to operate over a wider range of flow coefficients has been demonstrated, and any efficiency differences on design are expected to be slight. The high suction specific speed operational ability of the inducer type pump and the straight-through flow design results in much smaller, lighter, and faster turning designs than for other pump types. Consequently, axial-inducer pumps are more compact systems with lighter gearing. The installation diameter of an axial-inducer waterjet is only about 20 percent greater than its inlet diameter, while the installation diameter of a mixed-flow design is 65 to 85 percent greater than its inlet diameter. The smaller size is particularly important for installation in the restricted transom space available in the slender hulls of high-speed ships.

Axial-inducer type waterjets have been developed for and used on such ships as the Jetfoil, SES-100A, and PHM. The Jetfoil waterjets use single-stage axial inducers of 51 cm diameter that absorbed about 3.2 MW and were developed by Rocketdyne. Since 1987, waterjets for the Jetfoil have been supplied by Kawasaki Heavy Industries. In 1994 Kawasaki manufactured four KPJ-169A model single-stage axial flow, flush inlet waterjets rated at 5.24 MW. These waterjets were installed on the AMD 1500/Kawasaki Jetpiercer catamaran ferry *Hayabusa*. Kawaski offers axial flow waterjets with rated inputs up to 20.0 MW, but has not yet built any waterjets this powerful. The Marine Corps AAAV program uses 58 cm single-stage axial-inducer waterjets of about 1.0 MW for water propulsion as a current U.S. application of axial-inducer waterjet technology. These units were developed by Naval Surface Warfare Center, Carderock Division.

Aerojet Liquid Rocket Company developed the PHM units which had a power rating of 13.4 MW. The design was a two-stage, two-speed axial-inducer waterjet of nearly 117 cm diameter. The SES-100A also had two Aerojet two-stage, two-speed axial-inducer waterjets rated at 6 MW each, which gave that vehicle speeds approaching 80 knots. The second stage in each pump turned at much higher RPM and tip speeds than the initial inducer stage. The headrise produced by the initial inducer stage permitted running much higher tip speeds on the second stage. Second stage tip speeds in excess of 90 meters per second were used. The advantages of the higher speed second stage are that it can be made shorter to save space and weight, and the operating point for the second stage can be in a more favorable flow coefficient range for stage hydraulic efficiency. Gearing and shafting is more complicated for two-stage, two-speed pumps. Aerojet did extensive work for the 3KSES program on developing a 117 cm diameter, two-stage, single-speed axial-inducer pump of 30 MW intended for speeds of 90 knots. These units had inducer stage inlet tip speeds as high as 60 meters per second, with the high tip speed enabling reduced unit size. However, the 3KSES program was discontinued prior to full-scale testing of the hardware.

The power rating and design speed of existing and developmental waterjets is compared with the pumps needed for near-term small HS Naval ships in Figure 5.3.2-2. The figure shows that near-term power goals significantly exceed the power capacity of existing axial-flow waterjets. While some small HS Naval ship designs are compatible with mixed-flow waterjets, many near-term

designs require the reduced diameter of single-stage or multi-stage axial pumps to fit machinery in the slender hulls.

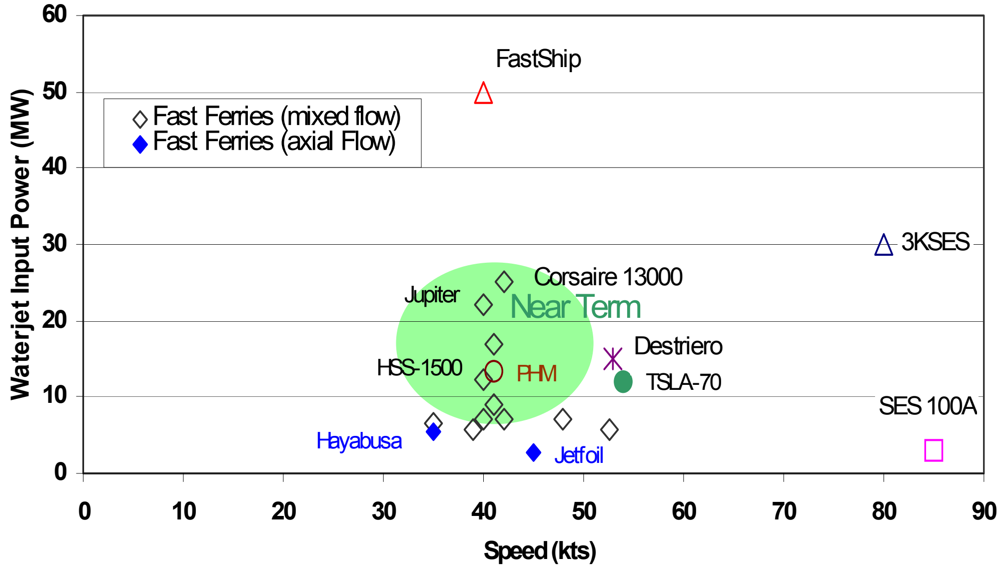

Figure 5.3.2-2: Waterjet Technology

5.3.3 Technology Goals

The technology to build waterjets that are matched to the relatively high-power gas turbines and high speeds of small HS Naval ships needs further development. The preferred configuration for small HS ships is one gas turbine for each waterjet. While demonstrated technology for mixed-flow waterjets is 25 to 26 MW, the demonstrated power of axial-flow pumps similar to those needed for HS ship applications has been limited to 13 MW. However, Kawasaki has a design for a 20 MW axial-flow waterjet. Moreover, much of the design and manufacturing technology supporting today's more powerful mixed-flow pumps is shared by axial-flow pumps. The near-term goal is to extend the capability to manufacture axial-flow waterjets to ship speeds up to 50 knots with applied powers of as much as 43 MW.

Single-stage axial-flow pumps are expected to be adequate for all displacement-type HS small Naval ship designs and some of the SES designs. However, two-stage axial inducers will be required for some SES designs where headrise requirements exceed the headrise ability of a single stage. This occurs for very high ship speeds and/or very high design point jet velocity ratios. Although two-speed, two-stage axial-flow waterjets have been built, single-speed two-stage pumps will be adequate for small HS ships. The additional blade row is expected to be a somewhat standard design that does not involve any undue mechanical complication. It would co-rotate with the first-stage inducer and be driven with the same shaft. The two-stage design reduces required tip speeds compared with a single-stage inducer design since the headrise per stage is reduced. However, for a comparable design point condition, both must pump a

comparable flow rate and the two-stage units provide only a slight reduction in inlet diameter. Thus, the two-stage axial-inducer design is considered an extension of the single-stage design.

Inlet design will require development for 2-3,000 mt, high-speed Naval ships. Inlet design must be matched to the geometric constraints of the specific hull design that is used. The long slender hulls favored for high ship speeds would limit the placement of inlets. Revised approaches to inlet design will need to be developed. Use of a single inlet to feed more than one waterjet is likely to be needed. The influence of hull boundary layer flow ingestion by the inlet has a large impact on waterjet performance. Waterjet inlets that maximize the capture of low-momentum boundary layer flow without adverse effects on ship drag will be favored. The impact of this low-momentum inlet flow can be as much as a 10-point improvement in the ship's overall propulsive performance coefficient. Ingestion of the lower-momentum boundary layer flow on the hull by the waterjet inlet will enable propulsive coefficients in the 70 percent range for very high ship design speeds. The nozzle location and orientation can generate additional lift by acting in a trim-tab like fashion to produce lift forces comparable to the total thrust of the jet. Understanding these effects and exploiting them to fully integrate hulls and propulsors offers significant potential for enhancing hydrodynamic performance. Inlet, hull, and jet interactions and inlet design are best studied and developed through the use of computational fluid dynamics (CFD) tools combined with model test data.

A better understanding of scaling effects between model and full scale is needed to reliably produce 2-3,000 mt waterjet-powered ships. Since boundary layers do not directly scale between model and full scale, the scale effects of hull boundary layers on the waterjet performance need to be carefully considered. CFD tools and model tests will play major roles in developing these important scaling relations.

Aeration and/or emergence of the waterjet inlet may result in a sudden and possibly severe drop in shaft torque that can have a serious impact on propulsion machinery. Hull model tests under different sea states are needed to predict and minimize aeration and emergence occurrences. Potential aeration impacts on shafting and the waterjet components and structure need to be considered in the design. Methods and systems to minimize aeration and/or inlet emergence, such as ride controls and wave sensors, need to be developed. Careful attention to stresses and waterjet bearings for these ocean-going Naval ships will be necessary to ensure long life.

If conventional jet deflection techniques are used, the relatively large powers and high ship speeds involved could require large, heavy steering and reversing structures that can weigh nearly as much as the waterjet itself. With a multiple waterjet arrangement, not all the units would require steering and reversing. Reversing may only be required at low speeds and powers, which would simplify the approach. This operational issue impacts the waterjet design and depends on the type of ship utilized. Steering can be effectively accomplished by means other than deflection of the jet, especially at high speeds. For example, high-speed ferries using "Interceptor" steering, a concept similar to an adjustable trim tab to impact hull flow, have demonstrated steering performance comparable to or better than that of ships equipped with steerable waterjets. In addition to reducing maintenance of waterjet steering gear machinery, alternatives such as Interceptors eliminate the loss of thrust that results from using waterjet

steering systems. Steering and reversing designs are critical elements of HS waterjet design in light of expected military mission maneuverability requirements.

Achieving the powering goals of up to 43 Megawatts will require waterjets to be developed that will absorb more than twice the power of today's most powerful axial-flow waterjets. This will require an understanding of the mechanical design and fabrication aspects of large waterjet designs as well as understanding pump hydrodynamics. Casting limitations on physically large pumps may make alternate approaches, such as fabrication with separate blades and components, a more realistic approach.

5.3.4 Overview of Development Plan

The development of waterjet technology is evolutionary in nature. As a result, the plan is focused initially on advancing the axial waterjet technology from today's 10-13 MW size pumps to the 43 MW near-term pumps.

The tasks, time to complete each task, and costs associated with developing the needed waterjet technology are shown in Figure 5.3.4-1. Two stages of waterjet development, model testing, and analysis are shown to address both near-term and far-term waterjet technology. Costs shown are engineering estimates, based on the expected scope of testing and facilities required. This hullform peculiar program will require essential data from other technology development efforts such as powering (section 3.2), seakeeping (section 3.3), and maneuvering (section 3.4), as well as model test data for hullforms of interest (section 2).

Years	1	2	3	4	5	Funding ($K)
Near Term Waterjet						
- Design & Analysis						1,500
- Enhance Analytical & Design Tools						600
- CFD Analysis inlet/hull/jet interactions						500
- CFD Scale Effect Predictions						250
- Inlet Design & Analysis						475
- Steering & Reversing Analysis						275
- Manufacturing/Fabrication Studies						150
- Stress Analysis						500
- Model & Component Tests						750
- Full Scale Development & Prototyping						9,000
Funding ($K)	1,375	2,475	5,300	4,850		14,000

Figure 5.3.4-1: Waterjet Technology Development Plan

Larger, higher power axial-inducer waterjets have not received much attention in recent times, but renewed development of this promising technology has high potential payoff. A near-term waterjet design for the 43 MW gas turbine is the next step in the state-of-the-art for axial-inducer waterjets. The technology needed to manufacture a pump optimized for the 40-50 knot design speed range will be produced. This represents an increase in axial-flow waterjet powering of

about 3 times the present demonstrated capability, and will result in a unit with an impeller diameter in the likely range of 2.5 to 3.5 meters, slightly larger than the 2.0 meter impellers currently being manufactured for mixed-flow pumps. A 4-year development cycle is required to produce the full-scale near-term pump prototype.

Steps in the development plan for the near-term 43 MW axial-inducer waterjet include:

1. Development would be aimed at a marinized gas turbine in the 40-50 MW power range that would be the likely power source. This power level favors ship designs having speeds in the 45-55 knot range and would represent a near-term technology for a very large waterjet.
2. Enhance analytical and design tools for the near-term speed and power range.
3. CFD analysis and prediction of inlet, hull, and jet interaction effects.
4. CFD development for full-scale design predictions and model to full-scale correlations.
5. Inlet analysis and design studies.
6. Steering and reversing gear analysis and design.
7. Mechanical design studies and manufacturing/fabrication analysis.
8. Scale-model testing to verify performance and cavitation scaling.
9. Full-scale design.
10. Stress analysis of all critical waterjet-related components.
11. Prototype testing on a ship of opportunity or installation on destination ship.

5.4 Reduction Gears

The weight of reduction gears can represent a significant portion of the total ship weight due to the high installed power required for high burst speeds. Technology development is required, as simply scaling-up existing designs to the desired power levels results in unacceptably high weights.

5.4.1 Introduction

HS ship designs use either a type of offset gear known as a locked-train double-reduction gear or epicyclic gears to transmit power from gas turbines to waterjets. The reduction gears are generally of the single-input, single-output type. All gears are non-reversing.

Offset gears are the most widely used type of gear for ship propulsion. These gears are well understood and available from numerous manufacturers. Offset gears are generally the lowest cost gearboxes due to the relatively simple machining and grinding required to produce the gear. The epicyclic gearbox, on the other hand, is not widely used at sea. The epicyclic gearbox provides a very high power density (i.e., it is smaller than the offset gear) and typically weighs less as well. The primary disadvantage of the epicyclic gear is its increased cost relative to the

offset gear. This is due primarily to the fact that the epicyclic gear contains more gear meshes than does the offset gear.

While conventional ships primarily use offset gears, when size and weight are important, as they are for small high-speed Naval ships, the cost disadvantages of the epicyclic gears become less of a factor. During the development of the near-term HS ship designs, both types of reduction gears were used. As a result, requirements for two separate development paths presently exist, as no gearbox of either type exists that fully satisfies the reduction gear requirements established by the designs.

5.4.2 State-of-the-Art

High speed ship designs used both offset (parallel shaft) gears and epicyclic gears. Monohull and trimaran designs used offset gears, SES designs used epicyclic gears, and catamaran designs used both offsets and epicyclics.

Both offset and epicyclic gears have been built in power levels as large as needed for a 40 to 60 knot Naval ship of 3,000 mt. While the largest power units have not been for marine propulsion, it is important that the manufacturing facilities already exist for such large units. For marine propulsion, larger offset gears exist than do for epicyclic gears. Offset gears up to 50 MW per shaft have seen service, such as those for aircraft carriers. While these designs are significantly heavier than required for high-speed Naval combatants, Philadelphia Gear has designed a lightweight offset gear of similar power for FastShip Atlantic that weighs about two-thirds the weight of the older design. With a weight of about 1.0 kg/kW, this represents the state-of-the-art for marine offset gears.

Epicyclic gears offer potential for significant size and weight reductions. However, the additional complexity of these gears translates into higher procurement costs. As a result, epicyclic gears have not seen widespread use for ship propulsion. The largest marine unit, a 21.3 MW gear made by the Swiss company Maag Gear AG to drive a booster waterjet, will go into service in early Spring 2003. This gear has a steel housing and weighs about 0.7 kg/kW. The largest epicyclic gear Maag has made is a 37MW gear for electric power generation at the Milan airport. Philadelphia Gear built a 90 kW epicyclic gear, with a weight of 0.88 kg/kW, for use in a hydroelectric plant. Cincinnati Gear designed a 30 MW epicyclic gear for the 3KSES program that had a weight of just 0.13 kg/kW. More recently, Cincinatti Gear offered a 25 MW epicyclic gear with a weight of about 0.2 kg/KW. However, Cincinatti Gear is no longer in business and these units all made significant use of aluminum for the housing. The design lifetime for the 3KSES gear was undoubtedly significantly shorter than that required for a Naval combatant. In any case, use of aluminum may not be feasible at power levels above about 30 MW. Potential weight gains due to the low density of aluminum are more than likely to be offset by the increased amount of material necessary to provide the required stiffness.

The 21.3 MW epicyclic gear built by Maag, with a weight of 0.53 kg/kW, is considered more appropriate to consider as the state-of-the-art.

5.4.3 Technology Goals

Near-term goals for both gearbox types are for lightweight designs capable of transmitting between 45 and 50 MW. Reduction gear weight is heavily influenced by the input power level, torque and reduction ratio. Figure 5.4.3-1 shows gearbox performance against power level for existing gears and HS ship designs. Performance is represented as weight per ‘torque’, where ‘torque’ is simply the power divided by the output rpm. The HS ship design points represent estimated design weights resulting from the final combination of gas turbine input power and speed and the design speed of the waterjet. Also shown on the curve are estimated trend lines for both offset and epicyclic gears.

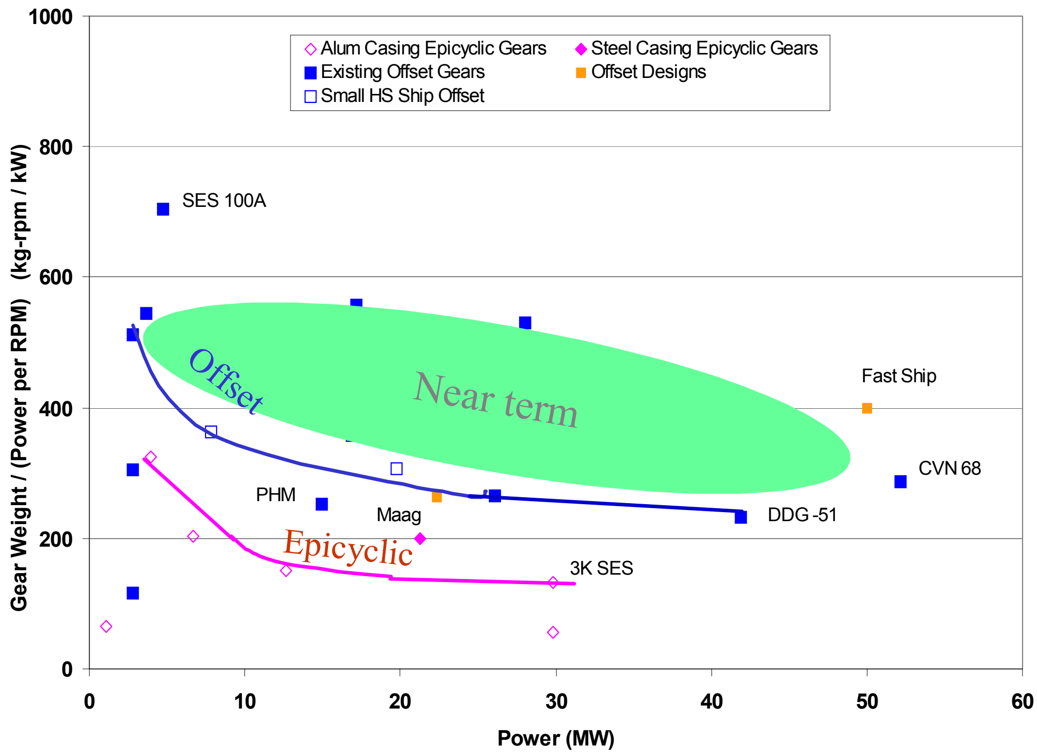

Figure 5.4.3-1: Reduction Gear Technology

In the near-term, the offset gears have weight per unit torque goals of between 250-400 kg/(kW/rpm), while the epicyclic gears have goals of 200-250 kg/(kW/rpm).

Figure 5.4.3-2 compares existing reduction gears with gears for HS Naval ship designs on the basis of weight per unit power. Offset gear near-term goals are 0.7-0.9 kg/kW. For epicyclic gears, the near-term goals are 0.5 -0.65 kg/kW.

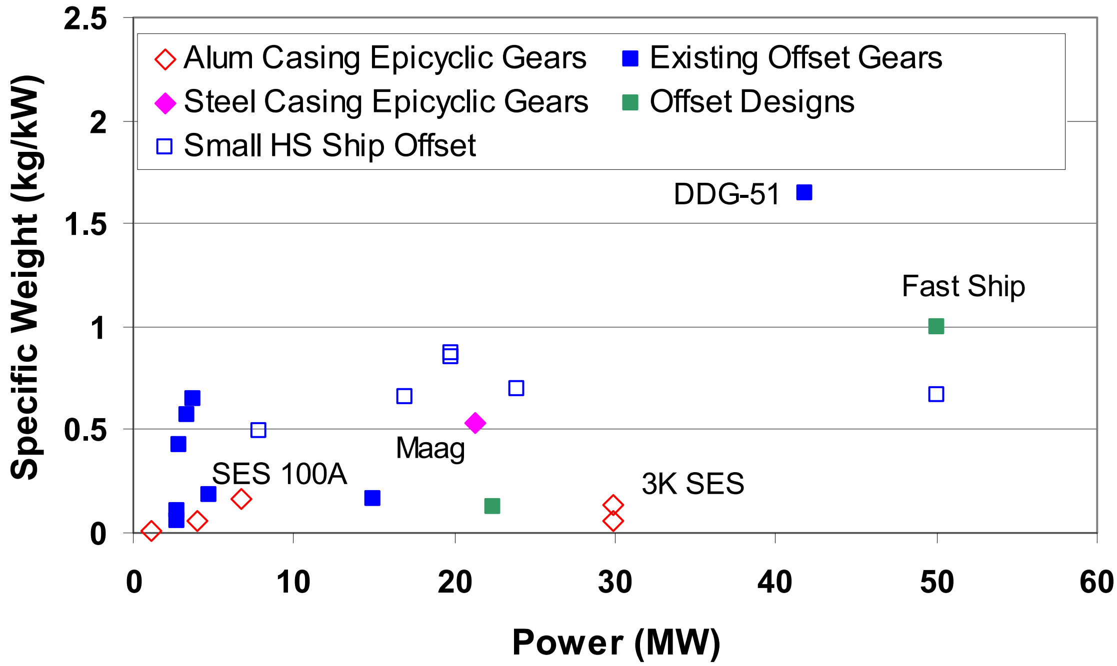

Figure 5.4.3-2: Reduction Gear Specific Weight

5.4.4 Overview of Development Plan

The principal focus for development for both offset and epicyclic marine gears is reducing the weight to the target levels. In terms of power output, similarly-sized gears have been built, if not for marine propulsion, at least for hydroelectric use. However, these existing designs are much heavier than what high-speed, small Naval ship designs require. Therefore, significant effort, principally in materials development, is required if the goals are to be met.

Traditionally, commercial ship reduction gear designs have been built with cost as one of the most important considerations, if not the most important consideration. Size and weight usually have less priority. Military gears have particular requirements levied on them that make low cost less important. But in both cases, weight is not usually an overriding factor. However, for small, fast Naval combatant ships to be technically feasible, the additional cost for lighter gears may be justified.

Major technology development in gear design optimization for minimum weight and materials improvements are required to achieve the target weight goals. Both offset and epicyclic gears will benefit from advances in these areas, although the payoffs may be different for each type.

Design optimization is principally a matter of investing the effort to make reduced weight the critical factor. For example, the required lifetime operating duty cycle and environment needs to be rigorously examined to determine whether design margins can be refined. Materials improvement relates to the use of stronger and lighter materials as well as the manufacturing processes required to make use of these materials possible. For example, operating at higher temperatures reduces cooling and lubrication requirements, but this requires high-strength steels able to operate continuously at these elevated temperatures.

In the past, improvements in machining accuracy have enabled lighter weight gears to be built. With more precision machining, tooth contact area is increased, reducing resultant stress levels. This has been translated into smaller and lighter gear meshes, reducing the overall gear weight. However, the current ability of gear manufacturers to produce gears of very high dimensional accuracy and surface tolerance is such that it is unlikely that any significant improvements are likely without potentially very large investment. Accordingly, the present plan does not account for any gains in this area.

Figure 5.4.4-1 presents a summary-level plan of technology required to achieve the specified near-term and far-term goals for both offset and epicyclic gears. Depending on the future Naval requirements for small high speed ships, it may be sufficient to develop only one type of reduction gear. There is a great deal of overlap in the development effort required; the primary differences only come into play when a specific design is being developed.

The plan shows that gears of both types can be available to meet the near-term goals in five years.

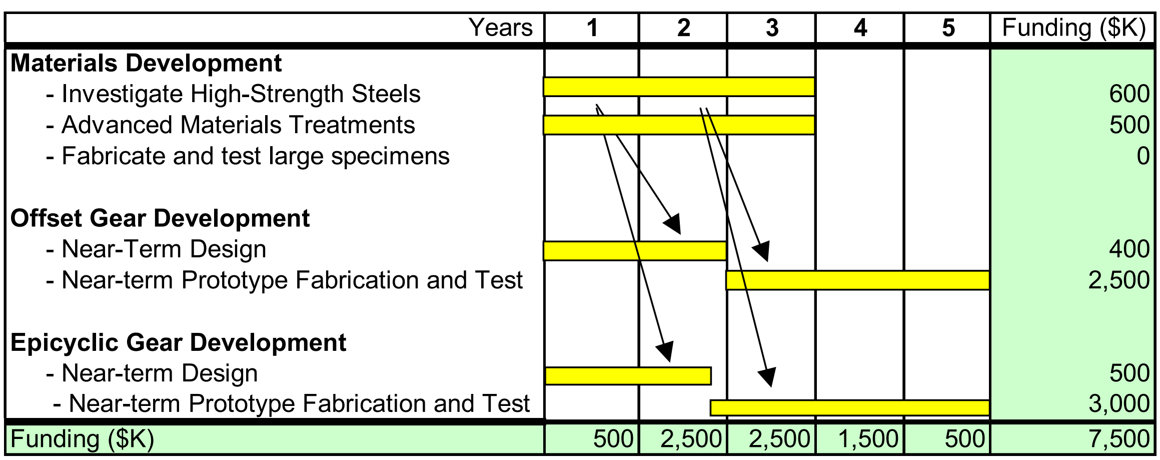

Figure 5.4.4-1: Reduction Gear Technology Development Plan

Development of advanced high-strength steels and associated post-forging treatments to increase the overall strength and durability of the materials used for the gear mesh is required. If successfully applied to large gears, weight reductions approaching 25 percent may be realized.

Large gears are typically fabricated using 9310 steel. There are several advanced high-strength steels that are becoming available and are in use for smaller gears. These steels, such as Carpenter's Pyrowear 53 and Vasco's X2-M, provide higher strength and the ability to operate at higher temperatures. These steels provide the potential for designing large gears that are inherently stronger than existing gears. Further, the ability to operate at higher temperatures reduces the requirement for cooling and lubrication, further reducing the overall size and weight of the gear mesh. This then contributes to reduced size and weight of the gear housing.

Advanced means of surface strengthening and finishing will be developed. Potential benefits include improved performance (higher strength), increased reliability and reduced costs. One such process is ausforming, developed by Penn State. Ausform finishing integrates heat treatment and hard finishing processes into a single phase. The process has been applied to much smaller gears than are necessary for high-speed Naval combatants, but the results to date show significant improvements in both final strength and dimensional accuracy. While none of the individual processes involved with ausforming are in themselves new to the industry, the integration of these processes into a single operation will require development to be applicable to large-scale gears. The near-term designs should be able to take advantage of the availability of reasonably large billets of the advanced high-strength steel.

Both offset and epicyclic reduction gears are being carried forward in this development plan. It may turn out that only one type of gearbox is ultimately required, but at the present time it is prudent to show development paths for both gears.

5.4.4.1 Offset Gear Development

The requirements for offset reduction gears may be satisfied by commercial development in the next few years, at least in terms of the power absorption requirement. If this is the case, it is likely that the near-term plan described herein will be unnecessary. While the weight of the commercially-developed gear may not be as low as desired, it may prove more cost effective to apply the funds targeted for near-term development towards the far-term requirements, where the weight goals are more critical and more demanding.

Present offset gear designs for large marine use have not been optimized for minimum weight. Since gear weight does not represent a very significant part of the total displacement, the shipbuilder will tend to opt for a lower cost gear. This means that the gear manufacturer will not devote a great deal of time to reducing weight. Doing so requires engineering development and that adds to the cost to the buyer, who is unlikely to pay for such extravagance. Further, to minimize life-cycle costs, the gears are typically designed to operate without failure for the life of the ship. This increases the size and weight of the gear, since design margins will be large to eliminate problems associated with fatigue.

As the speed of the ship increases, weight becomes more and more critical in every element of the ship. This allows the costs associated with reducing weight to be more acceptable. Detailed weight reduction engineering analyses, including comprehensive finite element analyses of the entire gearbox design, will be performed to identify potential weight savings. The impact of the lightweight designs on manufacturing costs will be assessed. Design margins will be analyzed to

determine where they may possibly be relaxed. It may be that reducing the design lifetime will provide weight savings, although achieving significant weight reductions via this path may require design lifetimes that are too short to be economically feasible. Further, the operating loads associated with very large, very fast ships operating in the open ocean will counteract to some extent the weight reductions available. Weight savings on the order of 20 percent may be possible through these approaches, with much of the weight reduction coming out of the housing (which may be as much as 50 percent of the total weight of an offset reduction gear).

The design of a prototype 43 MW gear will be developed. Representative design specifications (reduction ratio, operational and environmental loads, etc.) will be developed based on small HS Naval designs. The design effort will incorporate the results of the weight reduction analyses as well as the use of advanced steels, providing the results of that investigation indicate that sufficiently large billets of suitable material would be available in time to support the fabrication schedule.

Using the FastShip Atlantic offset gear as a starting point, it is expected that the combination of the systematic weight reduction efforts and the possible use of the advanced steel may result in a weight reduction on the order of 25 percent for near-term offset gears, leading to a weight of perhaps 0.75 kg/kW.

A prototype 43 MW offset reduction gear will be fabricated and tested under load by the manufacturer. The test setup will mimic the expected shafting and coupling arrangement between the turbine and the waterjet. The test will not accurately reproduce the full spectrum of loads associated with the at-sea application. Comparing strain gage and deflection measurements with those predicted for the test conditions will permit extrapolation to the expected at-sea loads. It may be feasible to consider an at-sea test of this gearbox, but this requires the availability of a ship powered by appropriately-sized turbines and waterjets. This is unlikely.

5.4.4.2 Epicyclic Gear Development

The development path for epicyclic gears is essentially the same as for offset gears. Therefore, only important differences between the two paths will be identified.

It is unlikely that commercial development will produce a 43 MW epicyclic gear in the next few years. Therefore, unlike the offset gear, the cost associated with the near-term epicyclic gear will need to be funded by the Government.

Much of the weight savings for epicyclic gears achievable through weight reduction design optimization come from the gear housing. Epicyclic gear housing weight is a smaller fraction of the total gear weight than for offset gears, so the potential weight savings are proportionately less. Weight savings for epicyclic gears through optimization are estimated at 15 percent at best.

The design of the 43 MW epicyclic gear assumes the use of the higher strength steels. Overall, the weight savings achievable in the near-term are expected to be on the order of 20 percent vice

perhaps 25 percent for offset gears. Epicyclic gears are generally somewhat lighter than offset gears of similar power, so a target weight of perhaps 0.75 kg/kW appears reasonable.

Prototype fabrication for the near-term epicyclic gear is essentially the same as for the offset gear. Costs are slightly higher simply due to the increased machining required to produce the sun and the planets compared to the offset gears.

The scope and effort of the prototype testing will be the same for the epicyclic gear as for the offset gear.

The gear technology development plan will produce very lightweight marine reduction gears that satisfy the requirements identified for HS ship designs. It is uncertain whether both or just one type will ultimately be required, especially with an eye towards the long-term, high-speed trans-ocean missions. Therefore, plans have been presented for both offset and epicyclic gears, as both have been identified as necessary to satisfy all of the HS ship designs.

The plan will lead to a high level of confidence in the final products since full-scale gears will be fabricated and tested. While the test programs stop short of testing installed in a ship, in-shop testing will permit confident extrapolation to at-sea conditions. Not fabricating and testing the prototype gears could reduce the total cost of the development plan. This approach is worthy of consideration principally for the near-term goals, as the weight targets assumed and the advances incorporated to achieve those goals are less demanding. However, full-scale prototype fabrication and testing is necessary to meet the more aggressive goals of the far-term gears. Analysis alone will not be sufficient to reduce risk to an acceptable level due to the combination of optimized weight designs, new materials, and new materials processing, combined with an operational scenario for which no previous design data exists.

Near-term commercial development for programs such as the proposed FastShip Atlantic project may produce offset reduction gears that come close to satisfying the near-term mission power transmission needs while falling short of weight reduction goals. Should this happen, the near-term offset gear development effort can be dropped. The additional weight gains achievable in the near-term would not justify the added expense. The related near-term materials work would still be required as it is an essential precursor of the far-term gear weight reduction effort

5.5 <u>SES Lift Fans</u>

5.5.1 Introduction

In addition to meeting performance requirements, SES lift fans must have as flat a pressure-flow characteristic as possible with a mild stall at low flow. These properties are necessary to allow the ship to benefit to the greatest extent from the SES concept.

5.5.2 State-of-the-Art

Lift fans suitable for a 500 to 3,000 mt SES are state-of-the-art. Fans meeting expected design pressure and flow requirements are commonly in use for industrial applications worldwide.

These fans typically have welded steel impellers and housings. In order to minimize weight, the fan impellers could be fabricated from riveted or bolted aluminum, and the housing could be of welded aluminum. For marine applications, the shafts would be of stainless steel. Fabrication in aluminum is available upon request. Advanced composite materials are used in the most recent SES fan systems and offer considerable advantages.

Lift fans required range from 2,000 KW in size up to 8,200 KW. The Navy's 3K SES design had six 2.18 m diameter centrifugal fans rated at 8,200 KW. Existing fans are of steel, but could be provided in aluminum with a weight savings of approximately 50 percent realized. Figures 5.5.2-1 and 5.5.2-2 compare state-of-the-art fan parameters.

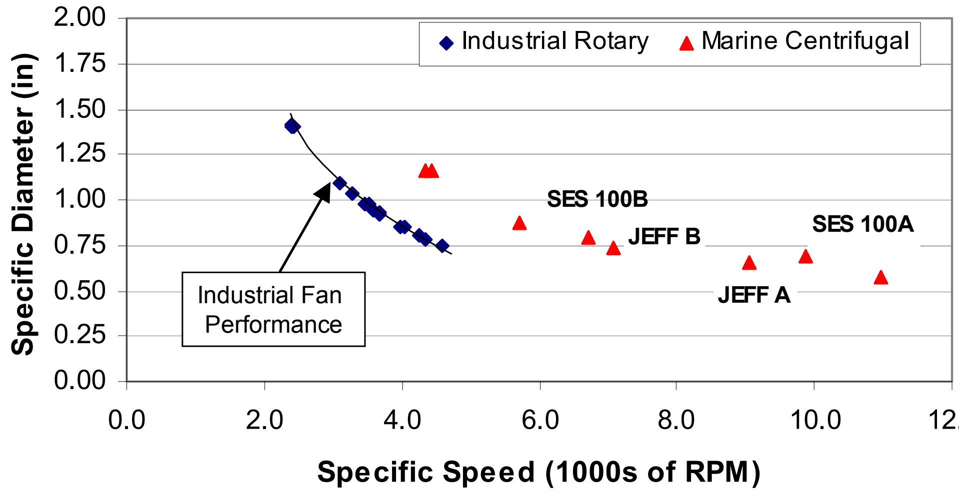

Figure 5.5.2-1. Comparison of Non-Dimensional Parameters of Existing Fans

The requirements for lift fans for SES up to 3,000 mt fall within the state-of-the-art for industrial centrifugal fans, and possibly for two-stage axial fans. Operation in a marine environment is not a major consideration for centrifugal fans. Some industrial fans are designed to operate in much more severe (corrosive) conditions. The use of composite materials for centrifugal lift fans has been amply demonstrated in a fleet of 9 SES MCM craft in Norway. Although the impeller diameter of these fans, designed by Band, Lavis & Assoc. in the U.S., was close to one meter, larger composite fans close to two meters diameter have also been built by UMOE in Norway. These BLA-designed fans have also been delivered to Finland where they have been installed in the Aker T2000 military ACV currently undergoing acceptance trials. UMOE states that they are able to design and build the same design in composite to four meters diameter. The composite SES fans have thousands of operating hours with perfect reliability and little maintenance. The UMOE fans incorporate a coating system which has completely protected the composite blades.

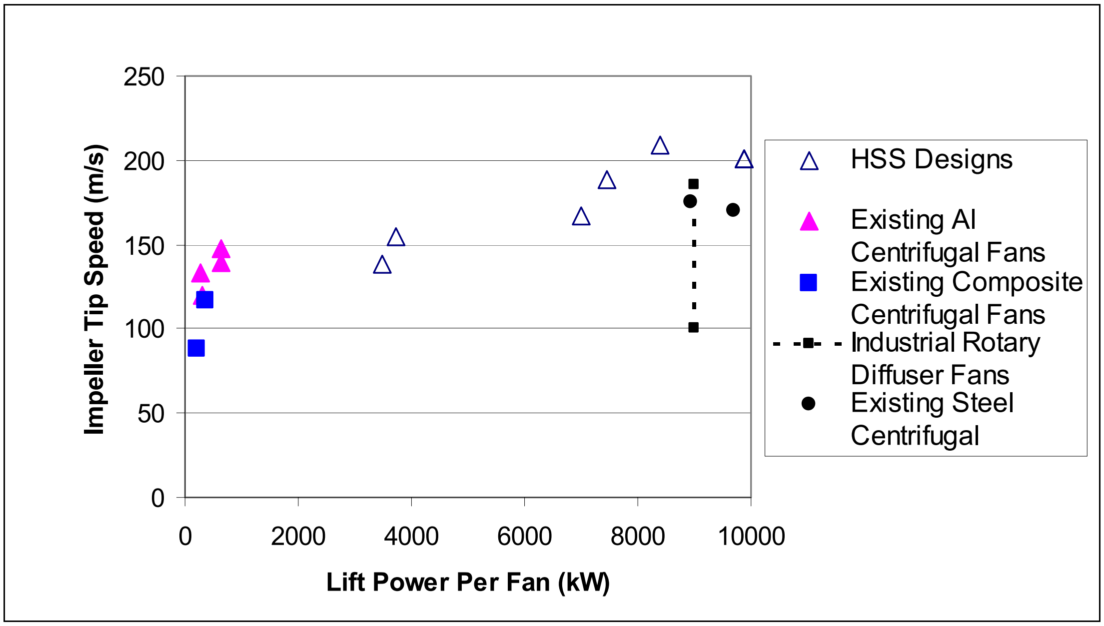

Figure 5.5.2-2. Comparison of "Power Density" of Existing and Design Fans

Of major concern to all fan manufacturers is the tip speed required to give the pressure required, which is about 2 m of water gauge, or 20 kPa., combined with the corresponding width of blade necessary to give the flow required. The combination of high tip speed and wide blade gives rise to high stresses that are difficult to meet in the structural design, especially when the stresses due to accelerations and gyroscopic loading from ship motions are added. This may be a limiting factor for steel and aluminum, or even titanium fans. However, the problem is much less severe for composite fans due to the lower density of composite materials.

Tip speeds of about 200 m/s appear to represent the state-of-the-art for centrifugal fan design. Fortunately, one of the most promising industrial fan candidates appears to have the lowest tip speed. Nonetheless, in this speed range, compressibility must be taken into account due to the high Mach numbers encountered.

Candidate manufacturers who have been contacted and have supplied information and data include: ABB, Aerophysics, Barron (a NYB company), and UMOE/BLA. Others who could not supply suitable fans include Chicago Blower, Howden Buffalo and Northern Blower. Aerophysics, at present, is able to design RD fans for SES that would then be made by selected manufacturers on a custom basis.

Fan development will need to focus on reducing weight through the use of lightweight materials and on marinizing commercial designs as required.

5.5.3 Technology Goals

The primary technology goal will be to reduce the weight of existing large industrial steel fans by use of lighter weight materials. A weight reduction goal of 50 percent is required to meet the weight value utilized in the designs. Prior experience with centrifugal fans has shown no issues with changing material from steel to aluminum. The manufacturers of the BLA-designed composite fans for the latest Norwegian SES Minehunter and Finnish ACV state that a carbon fiber/epoxy 3.7-meter fan is within their capabilities to manufacture. The dynamic blade loadings of the 3.7-meter design fan are only 80 percent of those of the Norwegian SES fans.

The near-term fan development goals are to refine the preliminary fan designs taking into account the marine environment. Weight and size of installation would be considered as well as the number of fans required. Drawings of the complete fan installation would be matched to the space available in the hull. In particular, the proposed UMOE/BLA composite fan design could be carried a step further.

Near-term goals are to:

- Seek the best combination of weight, space, power and performance.
- Pursue composite manufacture for very large fans.
- Refine the pressure and flow requirements.
- Study in more detail the fan performance characteristics during start-up and off-design operation.

In addition to total ship weight savings, lighter weight fans are desirable for reducing the gyroscopic loads on bearings and support structures and for reducing dynamic torque loads on the power train.

5.5.4 Overview of Development Plan

Existing commercial centrifugal lift fans can satisfy near-term technology objectives for small, high-speed Naval ships, but some development is required to decrease weight and improve reliability in a marine environment.

The primary development need is to investigate and validate the use of high-strength lightweight material alternatives, principally aluminum and composites. Other important issues to be investigated include acoustics, blade erosion and corrosion, and stall-free and mild pressure-flow characteristics at design and off design operations.

The size requirement for near-term fans is well within the state-of-the-art, and minimal development beyond building and testing a prototype is required.

The development schedule and cost for HS ship SES Lift Fan Technology is shown in Figure 5.5.4-1. Manufacturing costs provided by the fan suppliers are very similar for steel, aluminum, or composite.

Years	1	2	3	4	5	Funding ($K)
Near Term Lift Fan Development						
- Identify/Analyze Fan Alternatives						200
- Design Adaptation to Marine Environment						250
- Acoustic Studies and Tests						100
Fabricate/Test Near term Lift Fan						1,500
Funding ($K)	200	350	1,500			2,050

Figure 5.5.4-1. SES Lift Fan Technology Development Plan

5.6 SES Seals

Figure 2.5.1-1 compares the size and speed of existing SES with near and far-term HSS designs. The largest commercial SES built is the 56-knot, 1,500-mt Japanese TechnoSuperLiner (TSL) HISHO (KIBO) launched in 1994. The largest military SES is the 45-knot Russian Dergach 850-mt missile patrol craft. As is the case for all high-speed craft, both near and far-term HSS SES designs are substantially larger than vessels produced to date. While SES test craft have demonstrated higher speeds than those required for HS Naval missions, the combination of high speeds and larger sizes of the HS Naval designs place heavy demands on seal technology. Significant development is required to produce the technology needed to manufacture seals for HS combatant SES concepts.

5.6.1 State-of-the-Art

Through 1979, the emphasis on "high speed" for transoceanic SES such as 3KSES dominated seal technology development in the U.S. Subsequently, the emphasis has changed with the redirection of the U.S. Navy's SES technology studies towards the high L/B, slower craft (i.e. 35-55 knots). Currently, both high and low L/B SES are operating with simplified finger seal systems at the lower speed regimes with satisfactory results. The finger wear experience has been good, with over 1,000 hours of operation at speeds up to 55 knots and cushion pressures to 100 psf. The finger elements have always been considered frangible (i.e. remove and replace) items. Replacing the lower portion of the finger or cuff has proven to be an economic way of extending finger life. Typically, fingers can be removed and replaced without drydocking these craft.

European activity in the seal development area for both ACVs and SES has focused on improving the operational life of the bag and finger seals. Bag and finger seals were used exclusively by the British Hovercraft Corporation (BHC), one of the world's foremost manufacturers of ACVs, on their entire range of vehicles. This system has over 300,000 operational hours of experience since the basic version was introduced in the early sixties. The basic bag and finger design is also employed on the U.S. Navy's LCAC and the U.S. Army's LACV-30 amphibious lighter. Other versions of the design have been successfully employed on the SES-100B 100-mt high-speed test craft and on the British Vosper/Hovermarine HM-series of

SES. There were over 100 HM-series craft built and successfully operated over a wide range of commercial ferry applications worldwide.

In the mid-nineteen seventies the relationship between high over-water speed (60 to 85+ knots) and the wear rates of elastomer-coated-fabric fingers was not clear. Several component tests and full-scale environmental test programs were conducted during this period to obtain wear data. Full-scale environmental testing was conducted on candidate fingers for the 3KSES to determine wear rates. The dominant effect of over-water speed on elastomer-coated-fabric wear rates was readily apparent. While the effects of internal (cushion) pressure on finger wear was not fully characterized in these tests, the data indicated a lower rate of wear increase as a function of increased internal pressure as compared with speed induced wear. The overall evaluation of the bag and finger seal for the high-speed 3KSES application was satisfactory with respect to dynamic stability and performance, but unacceptable due to the high wear rates of the finger elements. It should be noted, however, that finger life at the lower speed and cushion pressure regimes, as clearly demonstrated by the heavy-duty commercial operations across the English Channel, were well within the acceptable range.

The experience of wear on the LCAC and LACV-30 skirt systems is based upon materials developed over twenty-two years ago. Understandably, finger wear rates on these craft tend to be high, primarily due to their abrasive, amphibious operations over concrete ramps, rough terrain, and ship well-deck transitions. It is noteworthy, however, that a relatively simple geometry change made to the LCAC skirt with the introduction of the LCAC Deep Skirt has more than doubled the finger life even though there has been no change to the material used to construct the skirt fingers. Significant advances have also been made over the past twenty years in the development of elastomer compounds for applications outside the SES and ACV areas. New, very lightweight material from a U.S. supplier has recently been tested at the labs at GKN Westland U.K. and underway on LCAC at ACU 5. Results to date are very encouraging. Further study of these technology improvements and collection of current operational data would be an important contribution to the seals design database.

Full-depth finger seals (i.e. the finger component minus the air supply bag) have been employed successfully on the BH110 series and, more recently, on Scandinavian SES, including the Norwegian MCM and Patrol Craft, the Norwegian Skjold SES patrol craft (which is capable of speeds in excess of 60 kts), and the Swedish SMYGE Patrol Craft.

Various SES designs have called for bow and/or stern seal retraction as an operational mode. Seal retraction is desirable for off-cushion, slow-speed operations to minimize overall craft drag and protect the seal from excessive hydrodynamic loading. The SES-100A in its original configuration was also designed for partial-cushion operations (i.e. side-hulls partially immersed) using retraction mechanisms for both bow and stern seals. Operational testing of this system aboard the SES-100A was not successful and the concept was abandoned.

5.6.2 Technology Goals

Bag and finger seals have evolved as the most reliable system within the context of current SES and ACV operations. However, it is recognized that operational parameters cannot be directly

scaled from current experience to HS Naval ship concepts without further development. With respect to the SES seals requirement, the major question that needs resolution is the ability of the seal elastomer-coated-fabric in immediate contact with the water surface to provide adequate operational life. At present, there is no adequate means of extrapolating seal wear (life) characteristics from existing data. There is clear evidence of elastomer-coated-fabric finger deterioration with increased over-water speed. In addition to the moderate to high-speed regime, the HS Naval mission requires operation at significantly higher cushion pressures. A better understanding of the effect of increased cushion pressure on finger wear rates is needed. An alternative seal configuration, such as the TMS seal, may reduce seal/surface flagellation and, hence, improve seal life. However, additional development of this system is required to resolve structural design issues before it can be seriously considered as a candidate seal system for the HS ship mission. The major thrust of the HS ship seals development plan must, therefore, be directed at resolving the finger wear rates and developing structural seal configurations to obtain acceptable operational life requirements.

The goal of the seal technology development effort is to produce the technology needed to manufacture bow, stern, and transverse seals to meet HS Naval SES performance objectives and provide acceptable reliability and maintainability qualities.

5.6.3 Overview of Development Plan

HS ship seal technology will be developed using a combination of analysis and testing. As the seals are essentially two-dimensional, the basic cross-section can be evaluated using simple force balance analysis based upon the defined cushion and seal pressures. Mathematical models of the primary seal components and their respective load paths will be employed to analyze maximum stress and load concentrations. Resistance of the system to hydrodynamic drag forces will be investigated by analytically deforming the seal (i.e. immersion) at maximum speed. The operating envelope will be evaluated for various combinations of seal components (e.g. the finger height and width, seal pressure). A key design consideration for the stern seal is rapid response to waves to minimize cushion leakage at the higher speeds. This requires careful design of the bag air supply and exhaust system to ensure that when the seal is compressed, the bag pressure is dissipated and then rapidly replenished to restore the seal to the nominal deployed position. Alternative designs for the stern seal will be examined.

Small-scale, static models of the seals will be used to verify the geometric proportions of the design and highlight problems. The scale-model test rig will include an air supply that is a scaled representation of the design bag and cushion pressure (not necessarily the flow rate).

A retraction system is needed for both bow and stern seals to minimize drag and potential damage to the seal during off-cushion operations. Retraction also allows for the easy replacement of sacrificial cuffs at the lower portions of the seals that see the greatest wear. Typically, bow seal retraction is more difficult because the seal (e.g. bag and finger type) is not conducive to full retraction. However, partial retraction of the seal is relatively straightforward as far as the multi-lobed bag is concerned. The full-finger seal, however, presents obvious problems with regard to retraction and requires further development. An operational representation of a retraction system designed for the 3KSES was employed on the modified SES-100A test craft.

While this planing seal had significant structural problems, the retraction system stood up quite well to operations.

Stern seal retraction has been successfully accomplished on several SES for both off-cushion operation and to increase side-hull immersion for ship pitch control and added waterjet inlet submergence to minimize air ingestion at certain operating conditions. Design consideration to account for "snatch" loads due to stern seal motions against the retraction straps in rough sea conditions is essential.

Applicability of a transverse seal will be determined by speed-power and motions requirements. If a transverse seal is required, the development process and retraction system would be the same as for the bow seal. The best transverse seal candidate is a bag and cone arrangement. The seal could be fed via a boost fan to increase the cushion pressure the required amount. An alternative seal type for the transverse seal is a full-finger seal.

The best candidate for the HS ship bow seal is a full-depth finger design having an upper and lower segment. The rationale for this design approach is based upon use of simple identical modular components that are readily maintainable or replaceable to extend operational life. Furthermore, there is no air supply required for this type of bow seal. An alternate bow seal design for HS missions that deserves consideration is the TMS type. The TMS seal requires further development to solve the problems of transverse modularity and lateral wave contouring. However, the superior performance characteristics predicted for this type of seal are worth pursuing further.

The best known stern seal candidate is a multi-lobed bag with wear strips at the seal/water interface. The stern seal is fed via a boost fan that takes cushion air and increases the pressure approximately 20 percent. The primary function of the stern seal is to minimize the cushion air leakage under all operating conditions. Consequently, the seal must be responsive to wave-forms passing through the cushion. These types of stern seals can be prone to flutter, wherein the velocity of the air passing under the trailing edge can cause an unsteady state at the surface, causing the seal to oscillate. Typically, this is more of a problem at model-scale and in steady-state (calm water) conditions. Flutter can be corrected by adding devices to the wear strip to break-up the air flow patterns. Alternative designs for the stern seal will be examined.

The plan for developing seal technology for the HS ship bow, stern and transverse (if applicable) seal systems is presented below. The tasks, time to complete each task, and costs associated with developing the needed seal technology are summarized in Figure 5.6.3-1. Seal technology development will focus on structural design, material requirements, maintainability, and producibility.

Time to complete (years)	1	2	3	4	5	Funding ($K)
Design & Development						500
Performance Verfication						1,100
Structural Design						500
Material Selection						1,100
Reliability & Maintainability						200
Producibility						200
Operational Verfication						3,400
Funding ($K)	1,400	2,700	2,900			7,000

Figure 5.6.3-1: SES Seal Technology Development Plan

The 3KSES database will be utilized as a foundation for the structural design approach. The structural design approach will consist of:

- a static loads analysis to define major load paths and stress concentrations of candidate seal designs subjected to a range of internal (seal/cushion) pressures and external hydrodynamic loads.
- a dynamic loads analysis using predictive techniques developed and correlated with test data under previous SES/ACV programs to analyze the effects of rapid seal redeployment (snap loads) and seal/cushion pressure transients for HS ship seals.
- a bow and stern seal structural design using existing 3KSES data as the basis for the seal structural design for the HS ship with appropriate selection and scaling of components and attachments to meet the seal system requirements. The structural design shall include the retraction mechanisms and all attachment fittings and hardware.

The material requirements analysis for the HS ship seals will include elastomer-coated-fabrics, attachment fittings and hardware, and retraction mechanism materials.

Analysis of the elastomer-coated-fabric materials is the most important area of study. Elastomer-coated-fabric materials technology developed under the 3KSES program will be extended to take advantage of technology developments over the past ten years.

Material requirements for the attachment fittings and retraction mechanisms are basically state-of-the-art and can be based upon a large database of previous testing and analysis. Slight extension of 3KSES technology is required to reflect operational experience with seal attachment methods and hardware developed under the AALC and LCAC programs and retraction mechanisms developed for the SES-100A and SES-100B to select materials and systems for HS ship concepts.

Candidate elastomer-coated-fabrics for HS Naval SES applications will be tested to fully characterize a seal material including:

- mechanical properties tests utilizing standard testing procedures (Federal Standard Methods) to verify the mechanical properties of candidate seal materials.
- small-ship environmental tests to test candidate material samples under simulated HS ship SES environmental conditions. Small-scale environmental tests are primarily used for comparison testing (i.e. known control seal material vs. candidate) to select the best candidates for further evaluation.
- large-scale environmental tests to simulate the full-scale operational environment of the HS ship SES seals, including internal pressure (cushion/seal) and external hydrodynamic loading.

Seal system structural design test requirements will be based upon a careful review of existing full-scale component test data and operational experience with both high-speed test craft and long-term commercial operations. HS bow and stern seal configurations will be tested aboard a sub-scale vehicle to verify attachment configurations and functional operation of the system.

Maintainability of the HS ship seals is a major design consideration, which will be addressed from the outset of this plan both in the design configuration and the structural design of the system. The seal system will be analyzed with respect to installation and removal of major components and dock-side repair capabilities. It is essential that maintainability be built into the seals design. The tasks conducted under this area will include studies of maintenance techniques and repair philosophy, and evaluation of the candidate seal systems. A Mean Time Between Failure (MTBF) analysis will be conducted for bow, stern and transverse (if applicable) seal systems. The analysis will be further broken down into the primary components of the seal (e.g. bag system, finger system, etc.) to determine failure modes and the overall life expectancy of the system. MTBF data will be developed from both operational experience (test craft, commercial experience) and environmental test data from the seal material test and evaluation activity. The maintenance cost of the seal systems will be a specific focus of attention under this plan. The need to “dry-dock” the ship for seal maintenance will be considered vs. dockside maintenance and the ability to maintain the seals internally (i.e. access through the wet-deck and plenum areas) while in the stowed or retracted position. The analysis will include a time, labor and services study on specific seal maintenance procedures with projected cost profiles.

Seal producibility is also a vital element in development of high speed SES seal technology. The capability of producing large runs of the selected HS ship seal materials while maintaining high standards of quality control will be assessed in an elastomer-coated-fabric produceability analysis. The processes and techniques of seam bonding will be evaluated with respect to reliability and cost. Seal attachment fittings producibility, centered on the primary attachments of the seal to the hull structure and the internal seal module-to-module attachments, will be analyzed. The candidate designs will be reviewed with respect to minimizing dissimilar components and for simplicity of manufacture. Finally, retraction mechanisms producibility will be analyzed to address the ease of manufacture and assembly of retraction mechanism designs and the integration of retraction systems with ship systems (i.e. power supply, mounting, control, etc.). As part of the producibility studies, the projected manufacturing cost estimates will be identified for the major seal subsystems with supporting rationale.

5.7 Machinery Systems References

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6.0 OTHER RELATED TECHNOLOGIES

6.1 Drag Reduction

Significant hullform technology development for displacement hulls and Surface Effect Ships (SES) is required to meet HS small Naval ship speed and range objectives. Displacement hull powering improvements are obtained through use of very slender hulls to significantly reduce wavemaking resistance. While wavemaking resistance is reduced, slender hulls have increased wetted area and increased frictional drag. Unlike lower speed ships, frictional drag of these slender hulls at 40 to 60 knots usually exceeds wavemaking drag by a considerable amount. This is not the case with SES, which derive their powering advantages through use of an air cushion to reduce hull wetted area, and hence frictional drag.

Many technologies have been proposed to reduce the frictional resistance due to the flow of water over a hull's surface. Investigations of these technologies have been conducted by many organizations over the past 30-40 years. While the physics of drag reduction have been demonstrated, none of the approaches have been found to be suitable for marine applications. Two of these technologies, which involve injection of drag reducing substances or micro air bubbles in boundary layers, have demonstrated significant frictional drag reducing potential in idealized laboratory experiments. Analytic models of the phenomena, backed by test data using laboratory-scale models (~6m in length) of simple shapes (e.g. flat plates), have been developed. Very significant frictional drag reduction has been demonstrated in these idealized conditions. Major technical issues remain such as the effectiveness of these technologies on complex ship-like shapes at ship scales in a marine environment, the amount of substance needed, and design of reliable, workable injection systems. Further basic research is needed to fully understand these technologies. Continued investment from outside the HS small Naval vessel arena is needed to determine their potential and practicality outside the laboratory.

If these technologies are scaleable to ship-like proportions, their potential for high-speed small Naval displacement ships is substantial, whether in the form of higher burst speed or a smaller, lighter and less costly propulsion system. The benefit is less for SES, due to the lower wetted area of these vessels while on cushion. The High Speed Sealift Technology Workshop experts concluded that development of suitable frictional drag-reducing systems would require more than ten years to produce viable ship systems. Consequently, these technologies fall well beyond near-term technology timeframe.

6.2 Composite Shafts

Power transmission shafts for ships are typically manufactured from steel. Steel shafts are heavy and require thru-life maintenance to counter galvanic and corrosion effects. While lightweight, non-metallic composite drive shafts are widely used in the aerospace industry, aerospace power/torque requirements are much lower than in the marine field. However, there is a limited marine market for composite shafting, particularly for high-speed ferries and fast patrol craft where weight is important. Composite shafts are in service in over 100 ships and craft at power

levels below 10 MW. Military applications include the Swedish SES test craft SMYGE (2.6 MW), Norwegian OSKOY class MCMVs (1.4 MW), and Norwegian Skjold (6 MW). Composite shafts for HS ship concepts must be capable of absorbing much higher power and torque.

The U.S. Navy has developed a mature technology to design and support fabrication of composite shafts for the range of powers and torques needed for HS small Naval ships. Technology development includes design, fabrication, and full-scale land-based tests of a 10 m long 37 MW/2,500 kNm shaft section including related couplings. By comparison, commercial state-of-the-art composite shafts are capable of absorbing about one-sixteenth the torque of this 37 MW shaft section. While full-scale at-sea demonstration of this shaft on a large Navy auxiliary (AOE) was planned in the early 1990s, the plan was not implemented. Development of high-power, high-torque composite shaft technology has continued to address critical issues such as shaft/coupling/bearing sleeve joints and shaft strength. These advances have included fabrication and full-scale land-based testing of additional high-torque shaft systems, Figure 6.2-1, to validate shaft to coupling joint strength, shaft to bearing sleeve joint strength, load sharing predictions, and predicted failure modes. However, in-service experience with high-power, high-torque composite shafts is minimal.

Figure 6.2-1: State-of-the-Art High-Power, High-Torque Composite Shaft

In addition to reduced weight, composite shafts offer a number of potential benefits to ships including reduced corrosion and galvanic effects, reduced magnetic signature, reduced bearing loads, increased vibration dampening, and lighter weight. While none of these attributes is critical to development of HS Naval ships, the significant weight reductions possible with composite shafts make this technology attractive for HS small Naval ship applications.

Shaft weight is a relatively small percentage of total ship weight in waterjet powered ships. Consequently, composite shaft technology is not considered essential to the viability of weight-sensitive HS ships. The relatively mature state of composite shaft technology, combined with the significant weight reduction possible, make composite shaft technology attractive to reduce empty ship weight of small HS Naval ships and increase payload capability. However, operational validation of this technology is needed to reduce risk to acceptable levels. Maximum power transmission requirements for small HS Naval vessels are approximately 43 MW/1,500 kNm per shaft. At-sea validation of a high-power, high-torque composite shaft is recommended since such a demonstration would be representative of near-term technology.

7.0 SUMMARY

This report fulfills one objective of the ONR-chartered High-Speed, Small Naval Vessel Innovation Cell project, which was to define near-term (available within 5 years) technology development requirements for 500 to 3,000 mt, high-speed Naval ships needed for realistic mission requirements. The requirements included burst speeds of 40-60 knots, payloads of 50-600 mt, and ranges of 3,000-4,000 n. miles at 18 to 20 knots. Since detailed design studies were considered beyond the scope of this project, whole-ship implications of technologies were investigated generically. A consistent set of standards was applied to all of the displacement hullforms considered: monohulls, catamaran, and trimarans. Technology projections for high-speed, small Naval ships were based mainly on near-term projections from the High-Speed Sealift Technology Workshop held at the Naval Surface Warfare Center, Carderock Division in October 1997. Projections for surface effect ships, or SES, were included. Where appropriate, these projections were updated to reflect recent developments. In addition to advanced hullforms, other technologies essential to these missions include advanced structures and materials, lightweight fuel-efficient gas turbines, reduction gears, waterjets, SES seals, and SES lift fans.

The capabilities needed from each of the technologies to produce HS small ships were compared with the technical state-of-the-art for those technologies to define the necessary near-term technology enhancements. Estimates of the time to develop and rough order of magnitude development costs were made for each of the technologies. The goal of this technology development effort is to bring the individual technologies to a level of maturity sufficient to lower risk to levels appropriate to ship design and construction. Technology development plans for each of the technologies were provided in earlier sections of this report. Figure 7-1 is a summary of those plans.

Year	1	2	3	4	5	Funding ($K)
Hull Form						
- Monohull						6,600
- Trimaran						7,900
- Catamaran						4,400
- Surface Effect Ship						6,100
Hull/Propulsor Integration						8,800
Structures & Materials						23,800
Gas Turbines						20,050
Waterjets						14,000
Reduction Gears						7,500
Lift Fans						2,050
Seals						7,000
Funding ($K)	9,200	35,275	42,075	18,750	2,900	108,200

Figure 7-1: Comprehensive Technology Development Plan

These development plans are comprehensive, with no allowance for market-driven technology development that may occur through commercial initiatives. Some technology development in

critical areas is expected to meet anticipated commercial needs. For example, development of larger gas turbines is highly likely for industrial, and commercial marine projects. While such commercial technology development efforts will potentially reduce the need for Government investment, elimination of this investment is not expected since there is some risk that the commercial efforts will either not come to fruition or the commercially-derived capabilities will fall short of the capabilities needed to meet the more demanding military requirements (e.g. duty cycle, sea state operability and structural loads, ambient air and water temperatures, maintenance philosophy). Consequently, the potential existence of these commercial efforts is identified, while the cost reductions that might result have not been shown.

The comprehensive plan shown in Figure 7-1 contains some necessary redundancies since the specific need for some of the technologies depends on other technology choices. The choice of hullform technology has a particularly large impact on requirements for other technologies. For example, development of near-term SES hulls requires development of SES-specific lift fan and seal technologies. Alternately, monohull and trimaran hulls may require development of different reduction gear technology than SES or catamarans. Hullform choice may be strongly influenced by mission parameters other than speed, range, and payload. Other characteristics, such as length, beam, and draft, vary considerably among the four hullforms considered. While most of the designs produced were within the required limits, significant military advantage may result from the differences in proportions of the hullforms. For example, the shallow draft possible with an SES on-cushion may prove compelling to expand port access. Since such decisions cannot be made with certainty prior to commitment to specific near-term objectives, the redundancies have been identified and retained at the individual technology level. However, it is unlikely that the full matrix of technologies will be developed. Choices between alternatives will likely be made to further focus the technology development effort and reduce cost. A hullform-specific plan for displacement hulls (monohull, catamaran and trimaran) is shown in Figure 7-2, while the SES plan is shown in Figure 7-3.

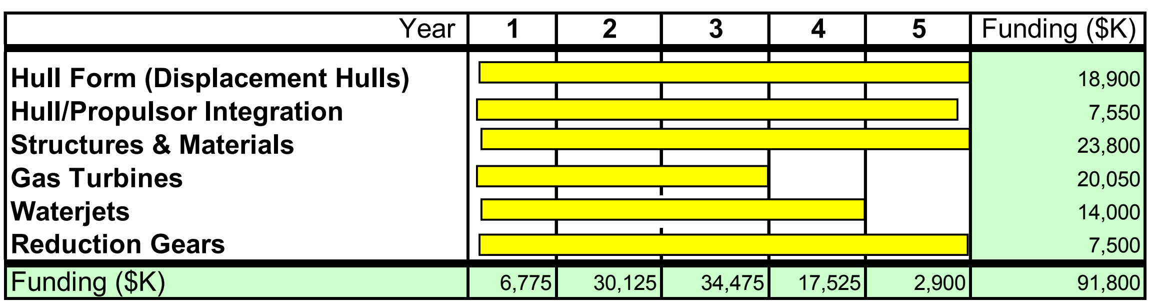

Figure 7-2: Comprehensive Technology Development Plan for Displacement Hulls

Several of the plans for development of individual technologies involve significant increases in scale from current technology levels. For example, near-term waterjets will require absorption of almost twice the power of today's largest waterjets. Similarly, near-term trimaran hulls would displace more than twice as much as the largest existing trimaran ship. Comparable increases in scale exist for advanced structures, gas turbines, reduction gears, and SES seals. Validation testing of large-scale specimens of these advanced technologies is included in the individual plans to validate the technologies, enhance technical credibility, and reduce technical risk to

levels suitable for ship construction. This validation testing can be accomplished for most of the technologies through land-based testing or at-sea testing in suitable existing or specially-

Year	1	2	3	4	5	Funding ($K)
Hull Form (Surface Effect Ship)						6,100
Hull/Propulsor Integration						5,050
Structures & Materials						23,800
Gas Turbines						20,050
Waterjets						14,000
Reduction Gears						7,500
Lift Fans						2,050
Seals						7,000
Funding ($K)	6,550	27,625	35,900	13,475	2,000	85,550

Figure 7-3: Comprehensive Technology Development Plan for SES Hulls

constructed ships. The costs associated with these large-scale tests are high. Costs associated with fabrication and testing of large-scale test articles accounts for all of the gas turbine costs, and 73 percent of the gear cost. Insertion of selected technologies such as lightweight structural components (interior decks, ramps, composite deckhouses), composite shafts, or reduction gears into design and build projects may reduce the R&D costs of these technologies, albeit at some increase in acquisition cost and programmatic risk.

The advanced hullform technologies needed to achieve perceived mission requirements can only be validated through construction and operation of large prototypes of the advanced hulls. Costs for building and testing advanced hullform demonstrators such as the RV *Triton* trimaran demonstrator, Figure 7-4, have not been included in this plan. The Triton project was focused on de-risking an advanced hullform for a 30-knot, 4,000-mt combatant mission. Validation of the hullform and structural technology was judged to be required at not less than 0.60 linear scale ratio and at speeds over 20 knots to reduce risk to acceptable levels. Construction cost of the 1,300-mt RV Triton was about $20,000,000 in 1998, while the two-year trials effort cost an additional $10,000,000. While not strictly applicable to HS ship technology development requirements, the Triton example is indicative of the level of effort required to validate hullform technologies for advanced concepts such as the slender HS displacement hulls or high L/B ratio SES.

This Technology Development Plan fulfills the goal of the 2002 High-Speed, Small Naval Vessel Innovation Cell project to define technology investments required to enable near-term development of 500 to 3,000 mt, high-speed Naval ships. For this project, high speed was defined as 40 to 60 knots. This Technology Development Plan defines the level of technology required, as well as the cost and time to develop, for the technologies essential to realization of high-speed, small Naval ships.

Figure 7-4: RV Triton, Trimaran Hull Technology Demonstrator

8.0 ACKNOWLEDGEMENTS

In producing this technology development plan for high-speed, small Naval ships we relied heavily on the technology projections incorporated into the High Speed Sealift Plan that was published by NSWCCD last year.

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